Basic Vibration Analysis Course 2031

World Headquarters
835 Innovation Drive
Knoxville, Tennessee 37932
Phone: (865) 675-3200
Fax: (865) 675-3205
www.mhm.assetweb.com

Emerson Process Management
Educational Services Center
12301 Research Blvd. - Building III
Austin, TX 78759
Phone: (865) 675-3200
Fax: (865) 675-3205

For information on training in the San Diego, California area:
Phone: (865) 675-3200, or Fax: (865)-675-3205

"ONE STEP IN YOUR JOURNEY TO BENCHMARK STATUS"

Copyright 2006, Emerson Process Management. All rights reserved.
Content for this manual provided by Emerson Process Management Training Instructor(s).

EMERSON
Process Management

Copyright 2006, Emerson Process Management. All rights reserved.

Basic Vibration Analysis Course 2031

This manual, as well as the software described in it, is furnished under license and may be used or copied only in accordance with the terms of such license. The content of this manual is furnished for informational use only, is subject to change without notice, and should not be construed as a commitment by Emerson Process Management. Emerson Process Management assumes no responsibility or liability for any errors or inaccuracies that may appear in this book.

Except as permitted by such license, no part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, recording, or otherwise, without the prior written permission of Emerson Process Management.

Please remember that existing artwork or images that you may desire to scan as a template for your new image may be protected under copyright law. The unauthorized incorporation of such artwork or images into your new work could be a violation of the rights of the author. Please be sure to obtain any permission required from such authors.

CSI logo, Infranalysis, InfraRoute, MachineView, Nspectr, Reliability-Based Maintenance and logo, UltraSpec, and WAVEPAK, PeakVue, RBM, RBMview, RBMware, RBMwizard, SonicS-can, SST, VibPro, VibBiew, are all registered trademarks of Emerson Process Management. Machinery Health is an impending trademark of Emerson Process Management. All other trade-marks are the property of their respective holders.

Written and designed at Emerson Process Management, 835 Innovation Drive, Knoxville, TN 37932, USA.

Brian Humes
VP & General Manager
Machinery Health Management
Asset Optimization Division
835 Innovation Drive
Knoxville, TN 37932

T (865) 675 2400 x2190
F (865) 218 1466
Brian.Humes@emersonprocess.com

Dear Emerson Process Management Training Customer,

We are pleased to have the opportunity to provide you training services from Emerson Process Management. The investment your company makes in technology and preventative maintenance systems can deliver value only when placed in the hands of trained and qualified personnel. By seeking continuous improvement through education and certification, you are taking an important step towards ensuring the long-term success of your plant's maintenance program.

It is our desire that your training experience at Emerson Process Management be valuable and personally rewarding. If you feel that any aspect of the training experience could be enhanced or otherwise improved, please let your instructor know at the end of your training session.

Sincerely,

Brian Humes
VP and General Manager

Contents

Chapter 1 • Introduction to Vibration

General Description · · · 1-2
FFT-Fast Fourier Transform · · · 1-4
Vibration Measurement Parameters · · · 1-5
Frequency Units · · · 1-7
Amplitude Units · · · 1-11
Amplitude Relationships · · · 1-12
Amplitude Conversion Formulas · · · 1-16
Phase · · · 1-17
Technical Components of Vibration Monitoring · · · 1-19
Review of Amplitude and Frequency Units · · · 1-26
Types of Transducers · · · 1-27
Accelerometer Mounting Response · · · 1-33
Signal Processing · · · 1-36
Problem Detection · · · 1-45
Transducer Location · · · 1-48
Machine Data Sheet · · · 1-49

Chapter 2 • Unbalance

Unbalance · · · 2-2
Case History #1 - Motor Driving Blower · · · 2-3
Case History #2 - Turbine Driving ID Fan · · · 2-6
Case History #3 - Coal Pulverizer · · · 2-9
Case History #4 - Reactor Fan #6 · · · 2-16
Case History #5 - Combustion Air Fan · · · 2-22

Chapter 3 • Misalignment

Misalignment · 3-2
Misalignment-Types and Descriptions · 3-4
Case History #1 - Line shaft Turbine · 3-6
Case History #2 - Axial Piston Pump · 3-11
Case History #3 - Centrifugal Air Compressor · 3-16
Case History #4 - Turbine Generator · 3-19
Case History #5 - Upper Quench Fan · 3-25

Chapter 4 • Mechanical Looseness

Mechanical Looseness · 4-2
Case History #1 - Pump Motor with Soft Foot · 4-3
Case History #2 - Torsional Looseness · 4-7
Case History #3 - Pump Driven by Motor · 4-11
Case History #4 - Vertical Pumps · 4-18
Case History #5 - Phase 1 Stack Fan · 4-19

Chapter 5 • Rolling Element Bearings

Rolling Element Bearings · 5-2
Bearing Fault Modes · 5-4
Fundamental Defect Frequencies · 5-5
Bearing Load Life Formulas · 5-7
Formulas for Approximating Unknown Bearings · · · · · · · · · · · · · · · · · · · 5-9
How Long Will the Bearing Last? · 5-10
Evaluating Failure Progression and Severity · 5-11
Analysis Parameters and Alarm Limits · 5-12
Typical Patterns of Normalized Bearing Frequencies · · · · · · · · · · · · · · · · 5-13
Antifriction Bearing · 5-14
Case History #1 - Tenter Zone Exhaust Fan · 5-15

Case History #2 - Primary Coarse Screen Reject Agitator · · · · · · · · · · · · 5-19
Case History #3 - Chemical Plant Sludge Pump · · · · · · · · · · · · · · · · · · 5-22
Case History #4 - Film Trim Takeaway Blower · · · · · · · · · · · · · · · · · · 5-28
Case History #5 - Paper Machine Press Roll Bearing · · · · · · · · · · · · · · · 5-31
Case History #6 - Reflux Pump North 2050 · 5-34
Case History #7 - Fan Pump · 5-36
Case History # 8 · 5-41
Case History #9 - Paper Machine Dryer Roll · 5-48
Case History #10 - Paper Machine Wire Return Roll · · · · · · · · · · · · · · · · 5-52
Case History #11 · 5-56
Case History #12 - Paper Machine Press Roll Bearing · · · · · · · · · · · · · · 5-60
Case History #13 - #1 Fire Water Pump · 5-63
Case History #14 - Inner Race Defect - #1 Ben Field Pump · · · · · · · · · · 5-68
Bearing ID Interpretation · 5-73
Bearing Interchange · 5-74

Chapter 6 • Gear Defects

Gear mesh · 6-2
Gear Ratio Calculation · 5-4
Calculating Gear Box Output Speed · 6-6
Gears · 6-8
Gear Signatures · 6-13
Gear Mesh · 6-14
Case History #1 · 6-19
Case History #2 · 6-22
Case History #3 - F.D. Fan #8 · 6-25
Case History #4 - Vacuum Pump Gear-Box · 6-27
Helpful information for successful gear box analysis · · · · · · · · · · · · · · · 6-30

Chapter 7 • Belt Defects

Belt Defects · · · 7-2
Case History #1 - Belt Driven Vacuum Fan · · · 7-3
Case History #2 - Forced Draft Fans · · · 7-8
Case History #3 - Belt driven over-hung fan · · · 7-15

Chapter 8 • Electrical Faults

Basic Electric Motor Construction · · · 8-2
Rotor Defects · · · 8-7
Case History #1 - Electrical Problem · · · 8-8
Case History #2 - Boiler Feed Pump Electrical Defect · · · 8-16
Case History #3 - Kiln Drive Motor - Electrical Defect · · · 8-21
Vibration Problems in Electrical Systems · · · 8-22
Glossary · · · 8-25

Chapter 9 • Journal Bearings

Journal Bearings · · · 9-2
Case History #1 - Direct Drive Centerhung Centrifugal Fan · · · 9-6
Case History #2 - Turbine Generator Set · · · 9-12
Case History #3 - Sleeve Bearing Looseness · · · 9-17

Chapter 10 • Resonance

Resonance · · · 10-2
Case History #1 - Reactor Fan #7 · · · 10-10
Case History #2 - DAF Pressure Pump · · · 10-14

Introduction to Vibration

Section 1

Objectives

- Define vibration.
- Describe the different methods of measuring vibration.
- Discuss the time and frequency domains.
- Examine amplitude measurements.
- Define the technical components of predictive maintenance.
- Determine the appropriate transducer.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

General Description

You can measure many different parameters for operating equipment - pressure, temperature, and flow, for example. However, of all the parameters you can measure, the vibration signature contains the most information. The vibration signature not only provides information concerning the severity of a problem, but it also points to the possible source of a problem.

Simply stated, ***vibration*** is a response to some form of excitation. The ***excitation*** is generally referred to as a ***forcing function***.

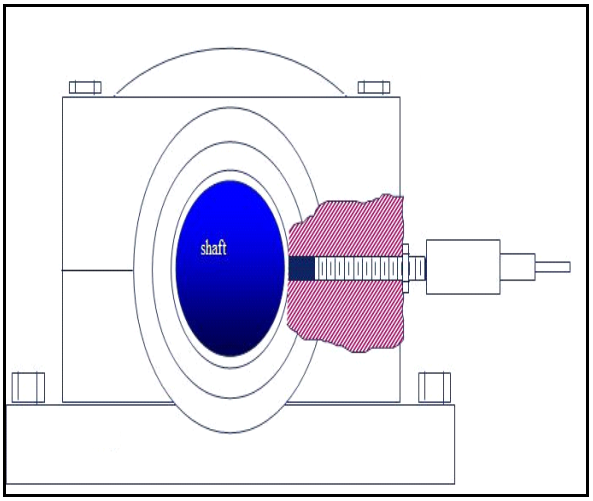


Figure 1

Figure 2

Figures 1 and 2 illustrate how vibration can be measured from a direct reading of the actual shaft movement within the case or from the casing of a rotating component.

Vibration can be observed in the **Time Domain** as the amount of time it takes to complete a particular cycle. In the illustration in Figure 3, the motion resembles a sine wave.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Figure 3 illustrates the movement of a machine. The overlying “PLOT” is a result of that movement. The waveform plot resembles a “SINE” wave.

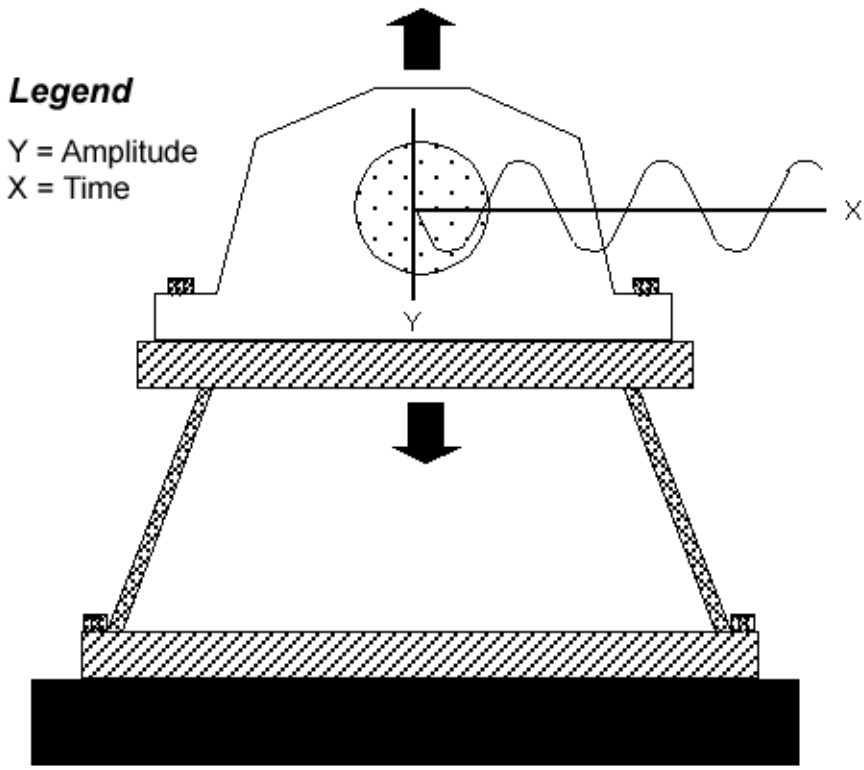


Figure 3

It should be noted that other components in or near the monitored equipment, such as belts, bearings, pumps, and fans in the equipment train will generate vibratory signatures. This energy can also appear in the data as additional signals. The resulting waveform may become very complex.

This complex waveform is transformed into a spectrum to be analyzed with respect to the frequency of various events. Most vibration analysis is performed in the spectral or frequency domain.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

FFT

The transition from time domain waveform to frequency domain spectrum is accomplished by the Fast Fourier Transform (FFT). A graphic depiction of the mathematical process is shown in Figure 4. The first plot (bottom left) shows a normal, complex time waveform. This complex time waveform is broken down into a series of individual sine waves, each one at a single frequency. As evident in the top graph, the individual sinewaves are plotted in a spread-out fashion. If the third plot is viewed from a different side angle rather than a front straight-on view, a new picture emerges. The final plot, on the right, shows "telephone pole" type peaks whose heights represent the sinewave amplitudes and the spacing on the horizontal frequency axis represents how often each event occurs.

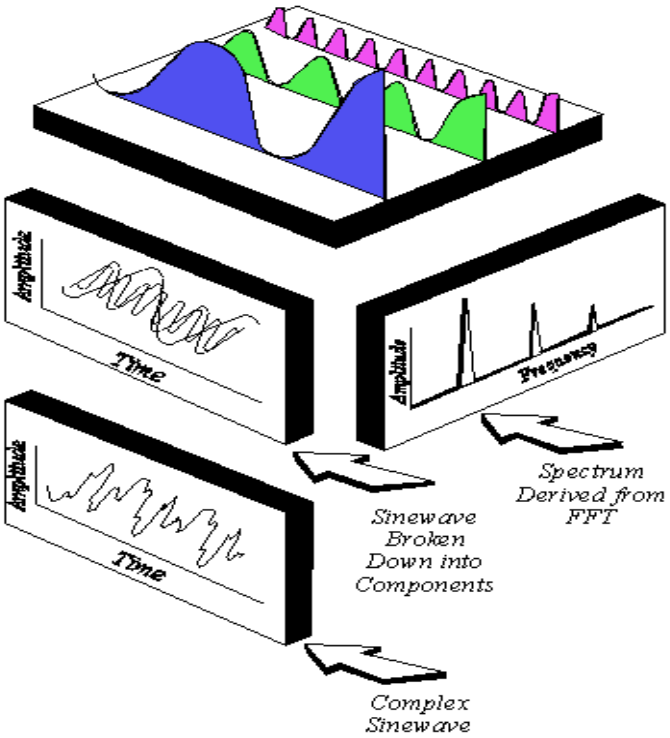


Figure 4

Fortunately, the spectrum analyzer performs the FFT process automatically at the push of a button and does not require that the mathematical calculations be performed manually. Remember that FFT refers to the process. Calling a spectrum an FFT is incorrect, although one may sometimes hear this term misused. Spectra is plural for spectrum.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Vibration Measurement Parameters

A vibration signal breaks down into two separate areas called domains. The **time domain** displays a plot called a waveform where the amplitude is displayed over time. For example, when an oscilloscope monitors an electrical signal, that signal appears in the time domain. The **frequency** domain displays amplitude as a function of how often an event occurs in some unit of time. An example of both domains appear in Figure 5.

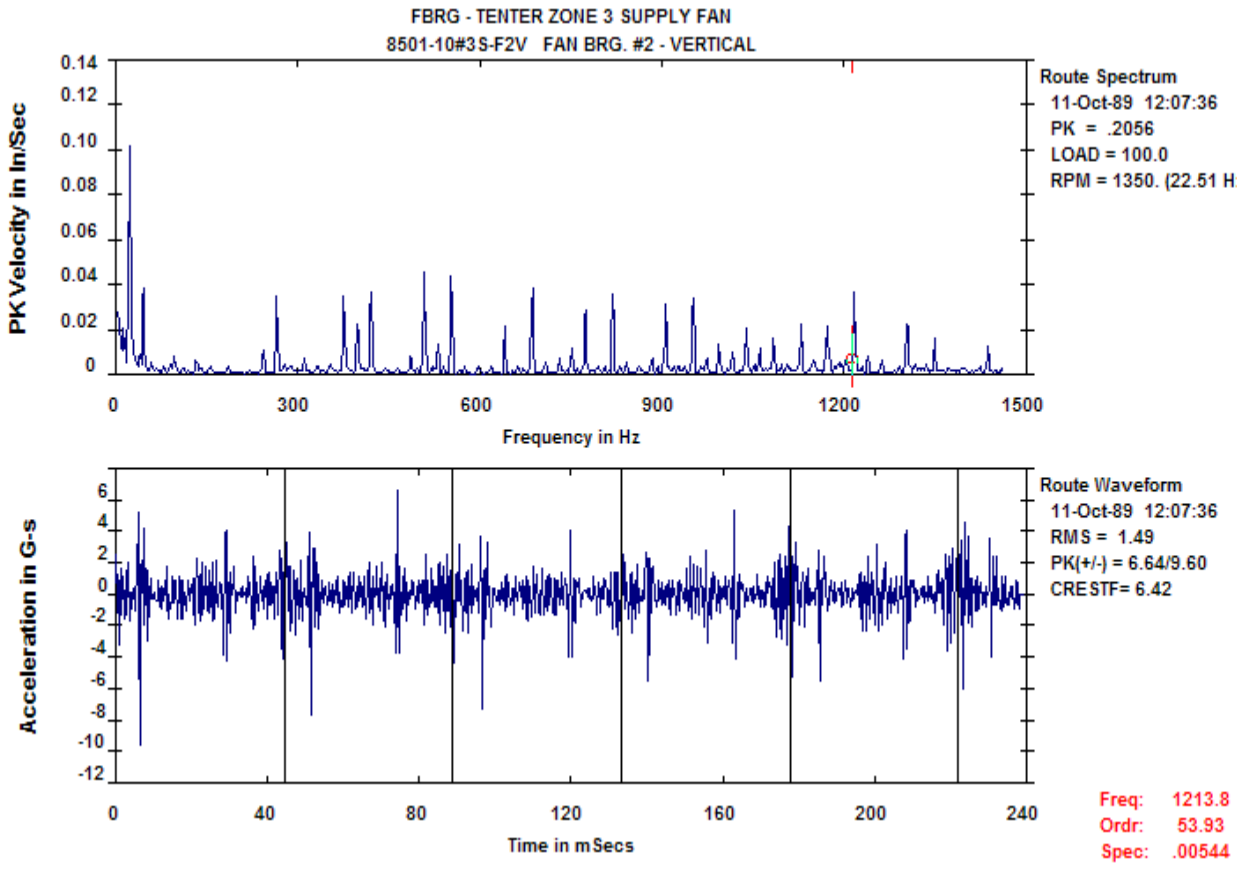


Figure 5

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The time waveform can help calculate a frequency. Establish a reference point in the waveform and then locate another point at some distance either to the left or right of the reference point. The time difference between the two points gives the DTIM.

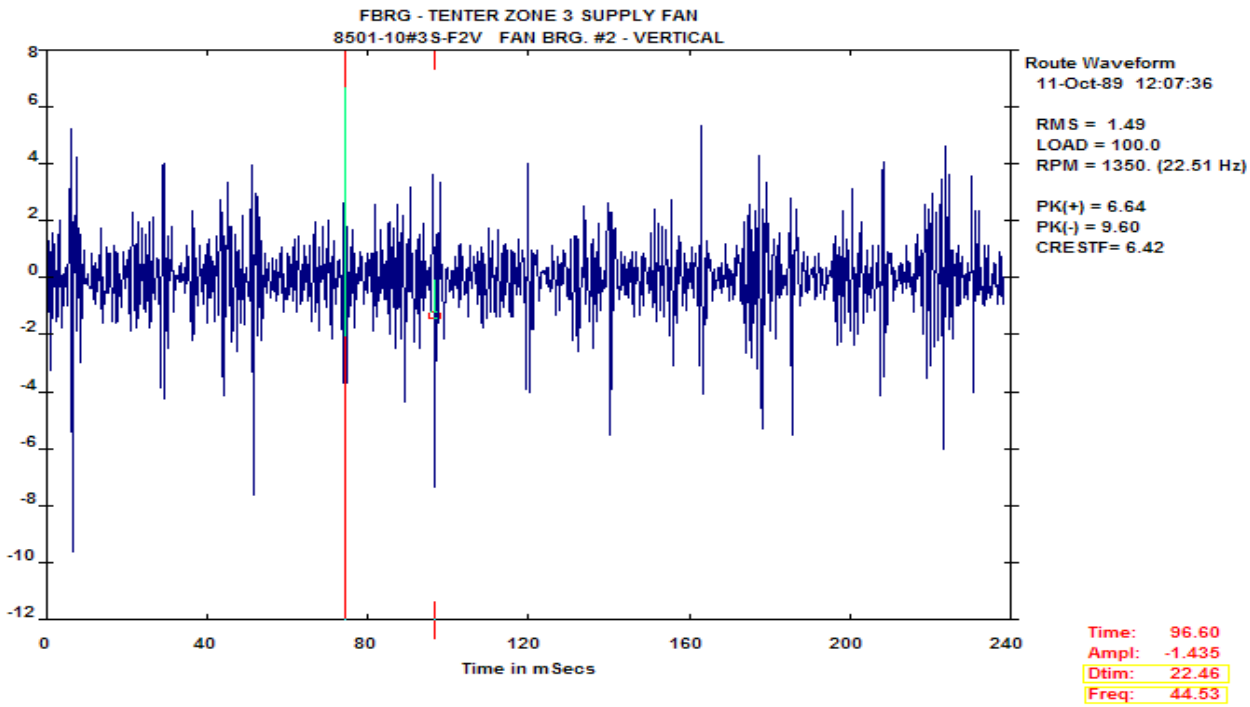


Figure 6

Because frequency is the inverse of the period (or time), frequency (f) can be expressed as 1 over DTIM (time difference) as illustrated in Equation 1. The units of time may be expressed as seconds, milliseconds, or as revolutions of the shaft.

Divide 22.46 milliseconds by 1000 to calculate seconds.

$$DTIM = 22.46\ milliseconds = 0.02246\ seconds$$

$$f = \frac{1}{T}$$

$$F = \frac{1}{0.02246}$$

$$f = 44.5Hz(2670CPM)$$

Equation 1

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Frequency Units

Frequency can be defined as how often an event occurs per unit time. For example, the bell in a clock tower chimes to indicate the time of day. It rings once for 1:00, twice for 2:00, and so forth, during a 24 hour period (one day). There would be 156 events, or 156 chimes per day. For someone who is paid once per month, that frequency would be once per month, or 12 events per year.

Similarly, for vibration data in the time domain, or waveform, units will be displayed as either time in seconds or revolutions. In the frequency domain, or spectrum, there are several choices as to how to display the units. The spectrum may be displayed in cycles per minute (CPM), cycles per second (CPS or Hz), or Orders, (units of shaft turning speed).

A vibration spectrum is displayed as an **X - Y plot**. **X** (horizontal) is the **frequency** axis, **Y** (vertical) is the **amplitude** axis. The X, or frequency axis, displays data with respect to how often a particular event occurs.

For example: a shaft is rotating at a frequency of 1785 revolutions per minute (CPM). It is also accurate to say that the shaft is rotating at a frequency of 28.75 cycles per second (CPS or Hz). Turning speed may also be referred to as one (1) order.

To convert any frequency from CPM to Hz, divide CPM by 60 since there are 60 seconds in one minute. To convert from Hz to CPM, ***multiply*** the value by 60.

For example:

1785 CPM / 60 = 29.75 Hz

Equation 2

29.75 x 60 = 1785 CPM

Equation 3

3550 CPM / 60 = 59.17 Hz

Equation 4

59.17 Hz = 3550 CPM

Equation 5

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

A pump may generate enough energy to appear in the vibration data. With a five-vane impeller, a 5-times turning speed signal is created. With every rotation of the shaft, five vanes pass any one point on the pump. As each vane passes, one event occurs. Since there are five vanes, five events occur per revolution. This is referred to as a 5xTS (5 times turning speed). Pump pass frequency is 5xTS. Multiply the turning speed of the shaft by the number of vanes on the impeller. The result is pump pass **frequency**. Other frequencies will be determined in the case histories presented in this manual.

The frequency domain displays amplitude as a function of how often an event occurs per unit time. The plot of amplitude versus frequency is called a spectrum and is illustrated in Figure 7.

A spectrum is usually displayed with peak velocity amplitude units on the vertical axis, while the horizontal axis can show frequency in hertz (cycles per second), cycles per minute (cpm), or orders (normalized to shaft turning speed). Spectra help analysts determine the machine defect or the source of a specific vibration signal.

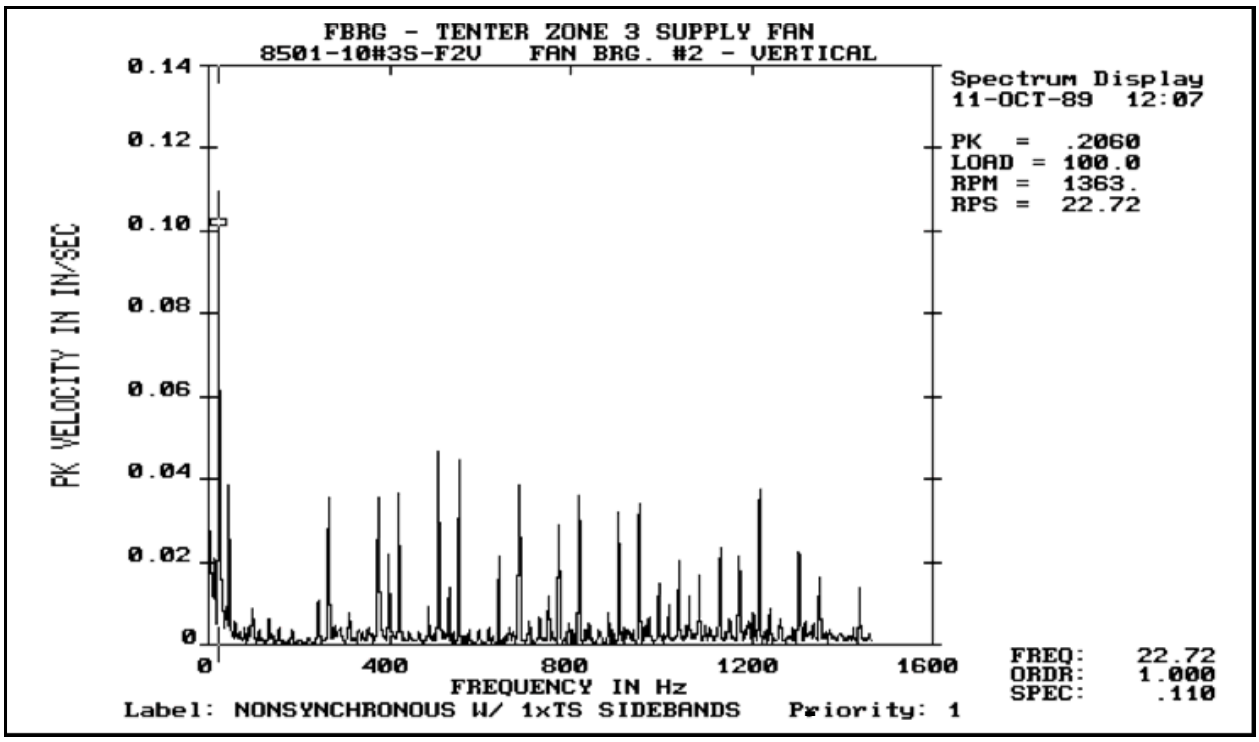


Figure 7

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Figures 8, 9, and 10 illustrate how viewing data in different frequency units has virtually no effect on the data itself. All the data is taken from the same machine but displayed in units of CPM, Hz, and Orders respectively.

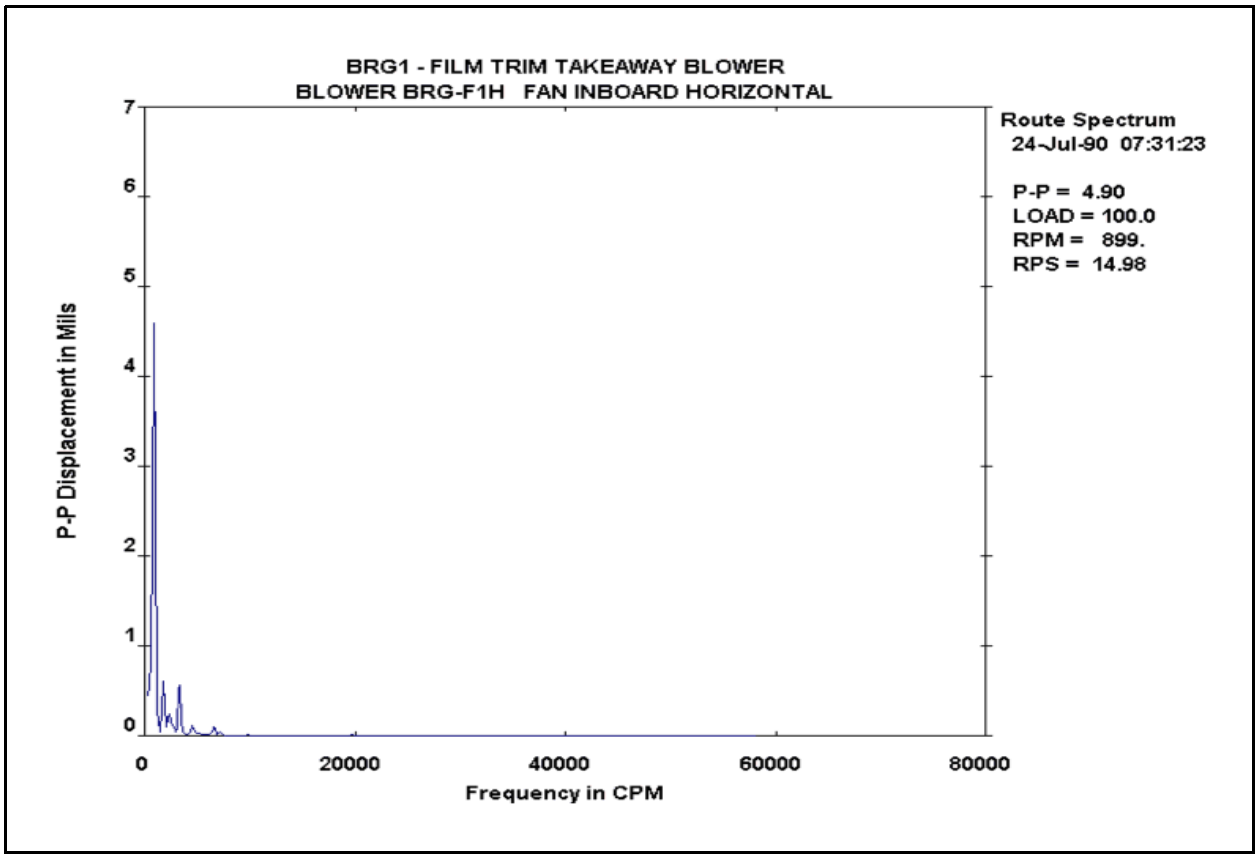


Figure 8: Data displayed in CPM

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

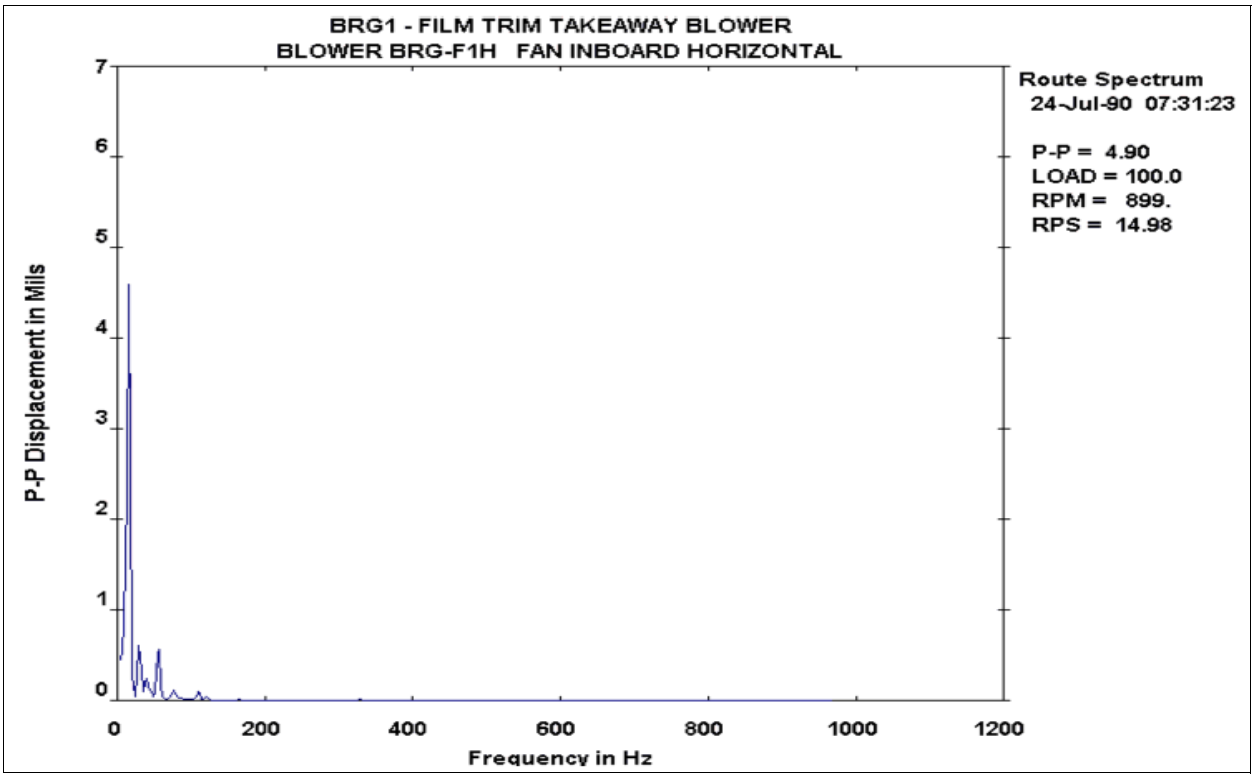


Figure 9: Data displayed in Hz

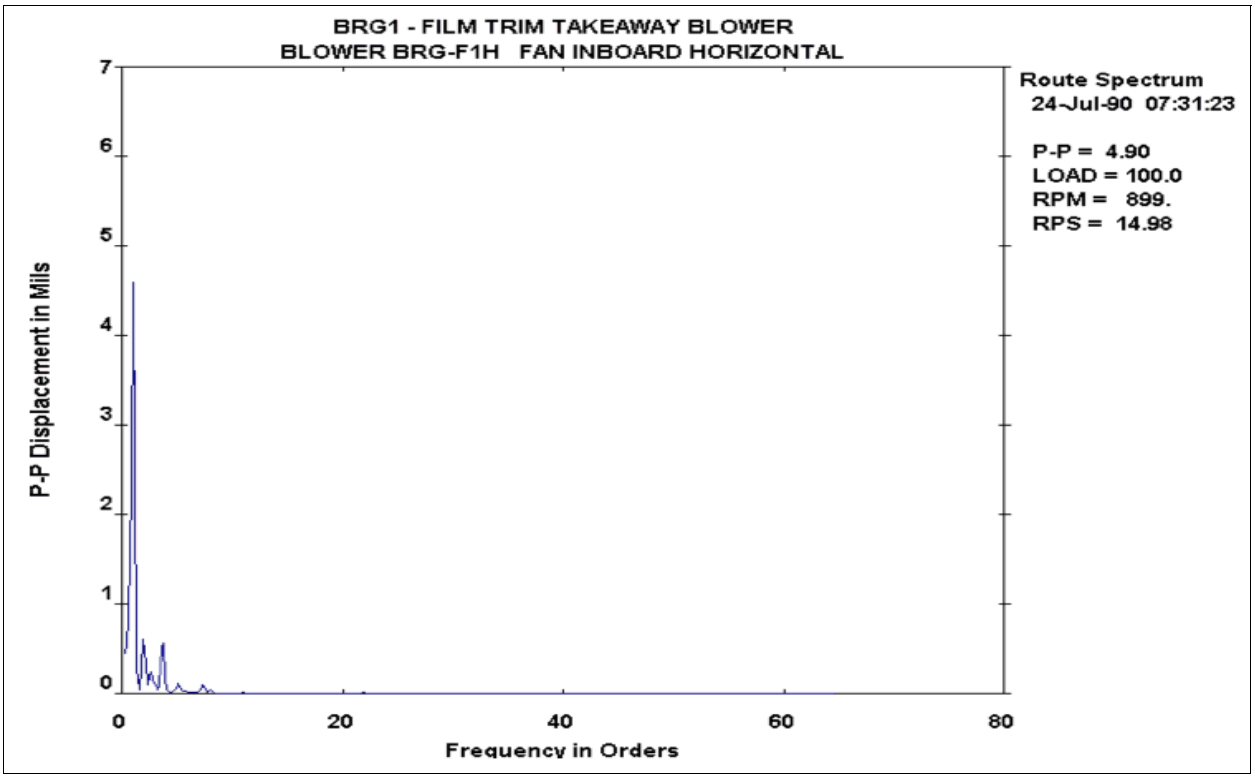


Figure 10: Data displayed in Orders

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Amplitude Units

The strength of the vibration signal is displayed as the amplitude in the time and frequency domains. Amplitude may be expressed in three units.

Displacement: total distance a body travels (*Peak to Peak*)

Velocity: the rate at which displacement occurs (*Peak*)

Acceleration: velocity per unit time; total force acting on a body (*rms*)

Displacement is commonly expressed in units of mils. One mil is equal to 0.001 inches

Velocity is commonly expressed in units of inches per second. (In./sec.)

Acceleration is expressed as units of force in G's. (1g = 386 inches per second^2)

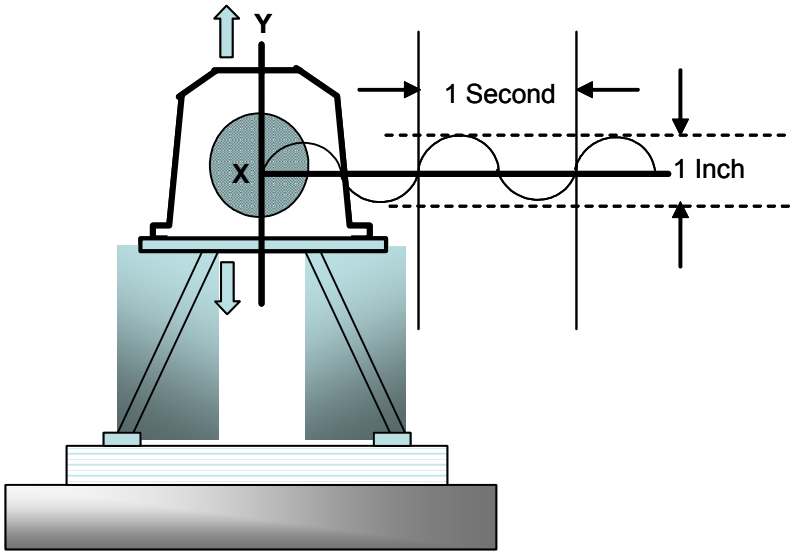


Figure 11

Displacement = 1 inch

Time expired = 1 second

Therefore velocity = 1"/sec

In an example of this event occurring at 87 Hz, the force required would be 1 g.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Amplitude Relationships

The three measurement types used to display amplitude are directly related to each other. For example, machines with a constant displacement experience a corresponding increase in amplitude for both acceleration and velocity as the frequency increases. Figure 13 depicts this relationship when one type is held constant. This information will help you determine which type of transducer to use for a given application

After the data is collected and transferred to the host computer, choose from three types of units in which to display the amplitude. Use either **0-to-Peak**, **Peak-to-Peak**, or **RMS**. The most common industrial applications are listed in Table 1.

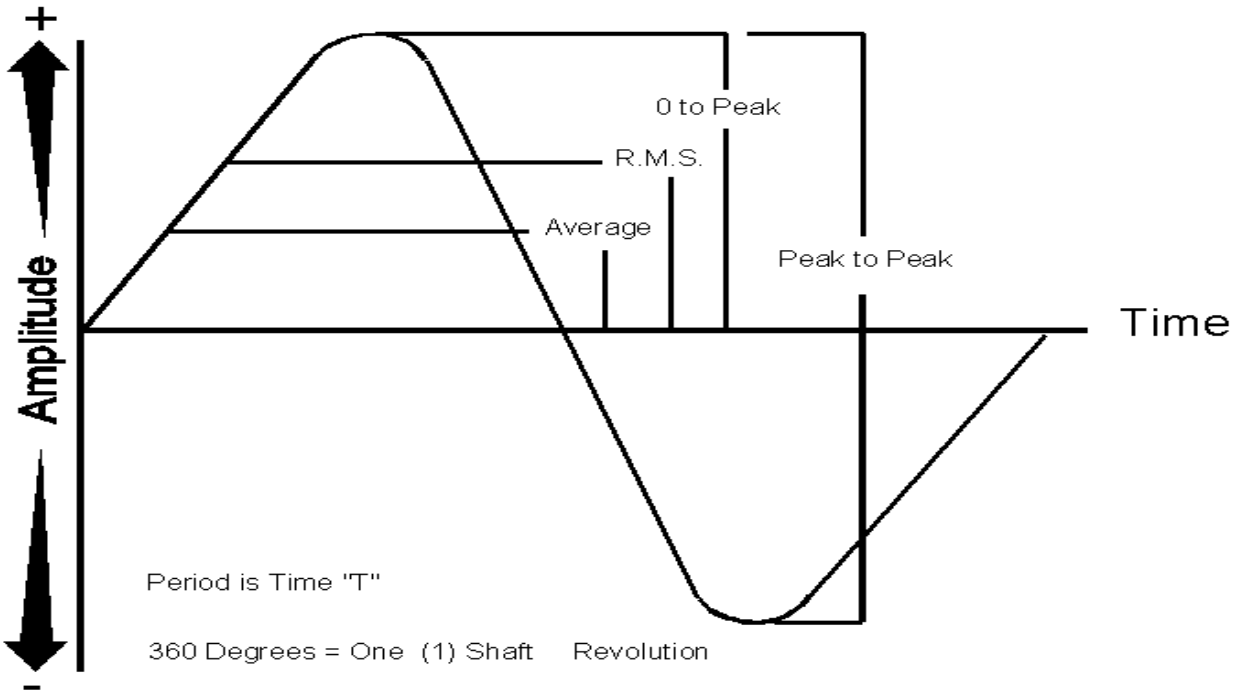


Figure 12

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Displacement	Mils	Peak-to-Peak
Velocity	In/Sec	Peak
Acceleration	G's	RMS

Table 1

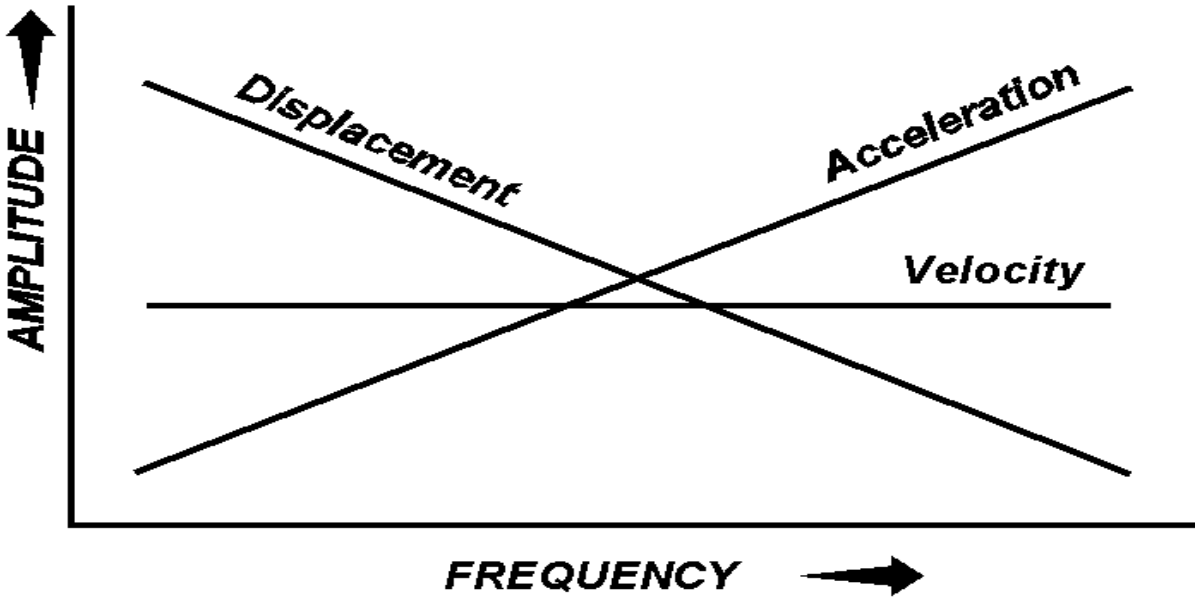


Figure 13

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The data in Figures 14, 15, and 16 illustrate the effect changing amplitude units has on spectral data. While most spectral analysis is done in amplitude units of peak velocity (see Figure 16), units of displacement are useful for detecting lower frequency events (see Figure 14). However, notice the significant increase in the peaks in the higher frequency range when viewing data in units of acceleration. Acceleration g's is useful in detecting early stage rolling element bearing defects (see Figure 15).

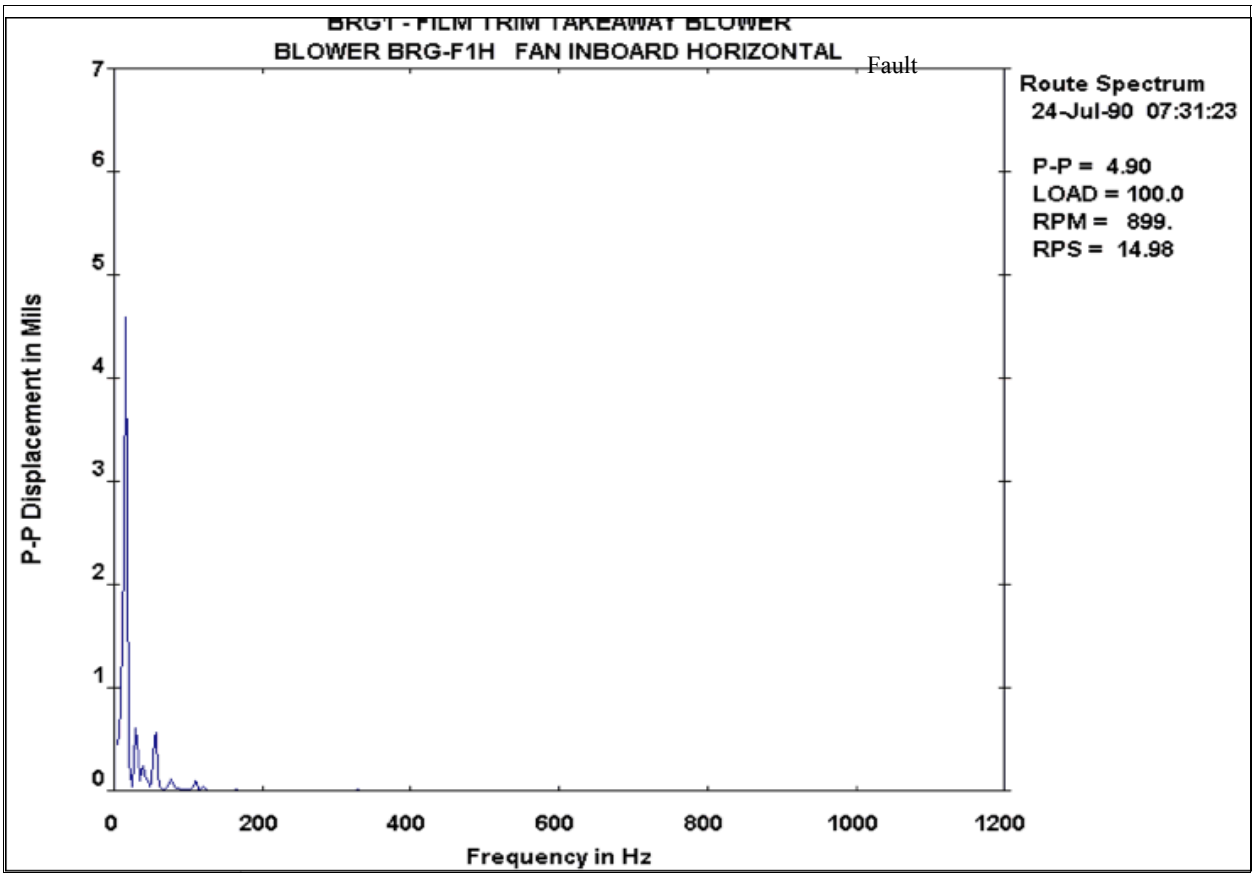


Figure 14: Displacement in Mils

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

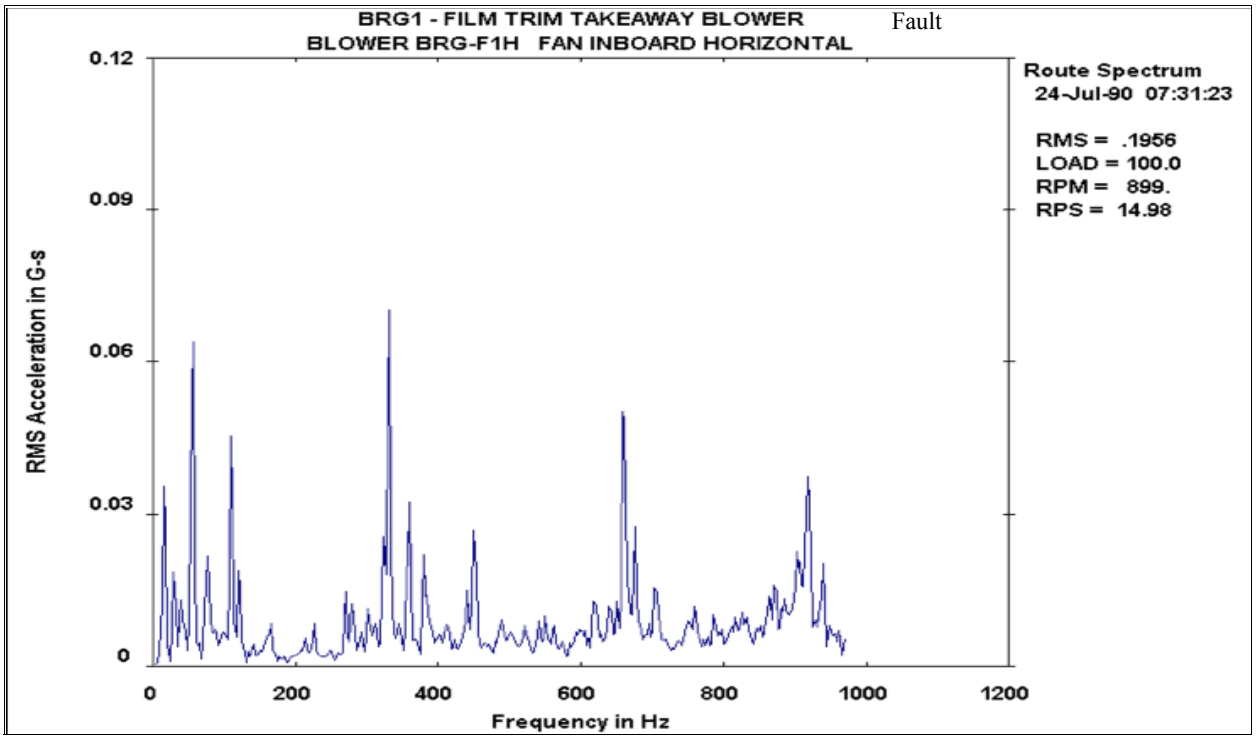


Figure 15: Acceleration in G's

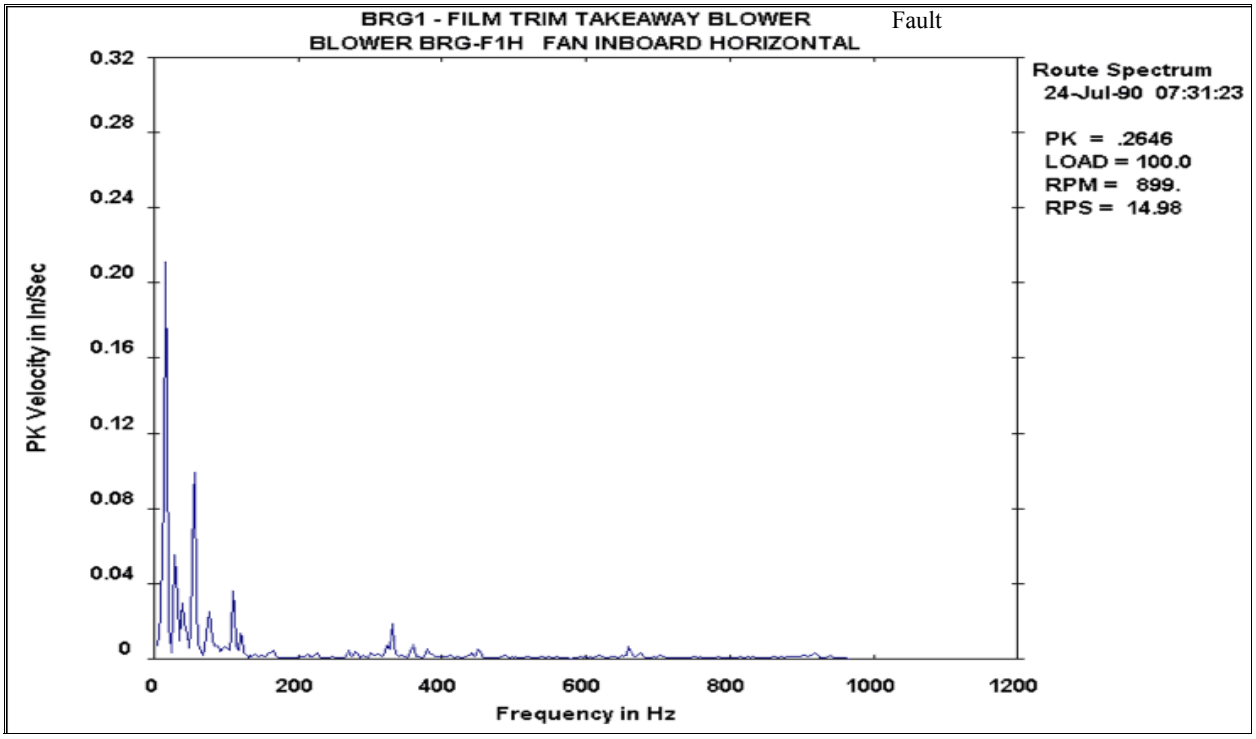


Figure 16: Velocity in In/Sec

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Amplitude Conversion Formulas

Amplitude is the measurement of the energy or movement of a vibrating object. The change in amplitude corresponds with the change in the severity of the problem. Conversion factors for the three units of amplitude are shown below in Table 2

.

RMS	Root Mean Square	0.707 times the true peak value
A	Average	0.637 times the true peak value
PK-PK	Peak-to-Peak	2 times the true peak value
PK	Peak	1.414 times the rms value

Table 2

Amplitudes may be mathematically converted from one unit to the other using the correct equations under certain conditions. These equations are **frequency specific** and must be applied to **sinusoidal** waves only. They are not intended for converted overall amplitudes.

$$V = 0.0031416 \cdot f \cdot D$$

$$A = 0.01146 \cdot V \cdot f$$

$$A = 0.00003613 \cdot D \cdot f^2$$

$$D = (318.47 \cdot V) \div f$$

$$D = (27,668 \cdot A) \div f^2$$

$$V = 86.75 \cdot (A \div f)$$

Reminder

These equations are **FREQUENCY SPECIFIC**. They must **NOT** be used to convert overall amplitudes.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Phase

Phase is the relationship between two events (comparing a phase reference pulse to the next positive peak of the vibration signal). Phase is measured in degrees of rotation or radians. Emerson Process Management's CSI equipment measures phase as ***phase lag*** - the interval from the phase pulse to the positive vibration pulse.

In Figure 17, the heavy spot on disk C passes by the transducer 270° after the phototach triggers. The phase lag of the system is 270°. Most digital analyzers measure phase in this manner. Analog machines measure ***phase lead*** - the opposite of phase lag.

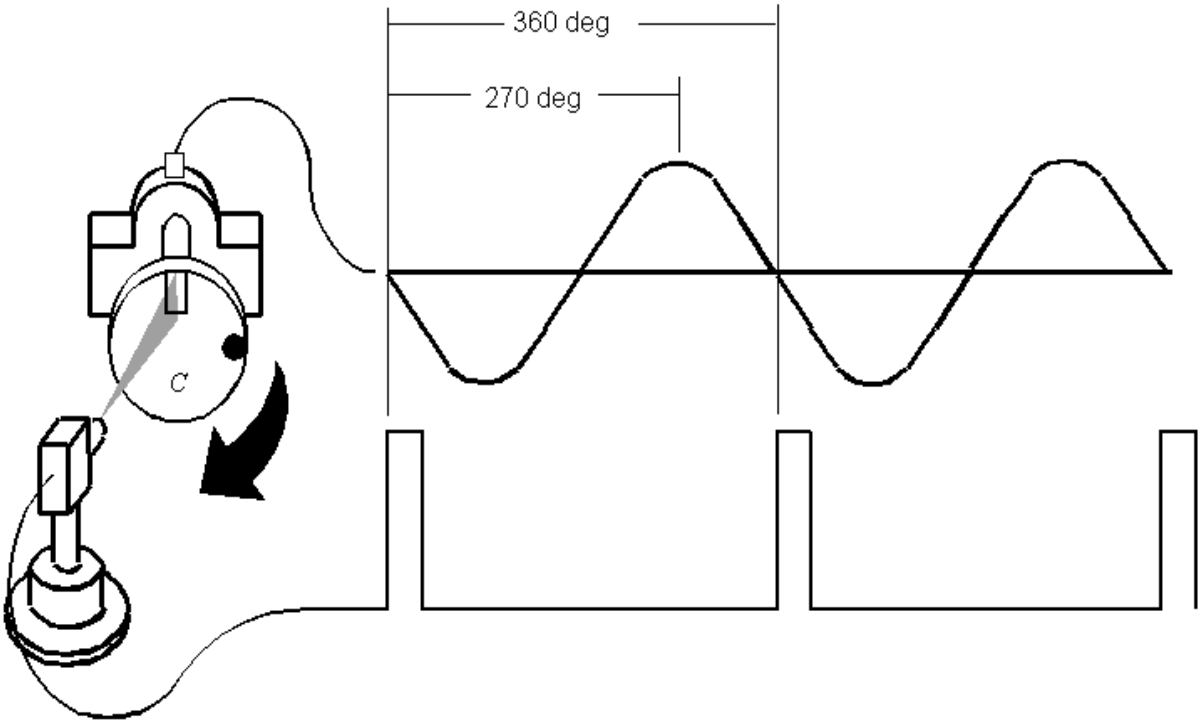


Figure 17

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Phase data may also be used to describe the relationship between the vibratory high spots on two rotating elements as illustrated in Figure 18. The heavy spot on Disk A is 180° out of phase with the heavy spot on Disk B. Disk B is generating a higher amplitude, or stronger signal, due to greater mass.

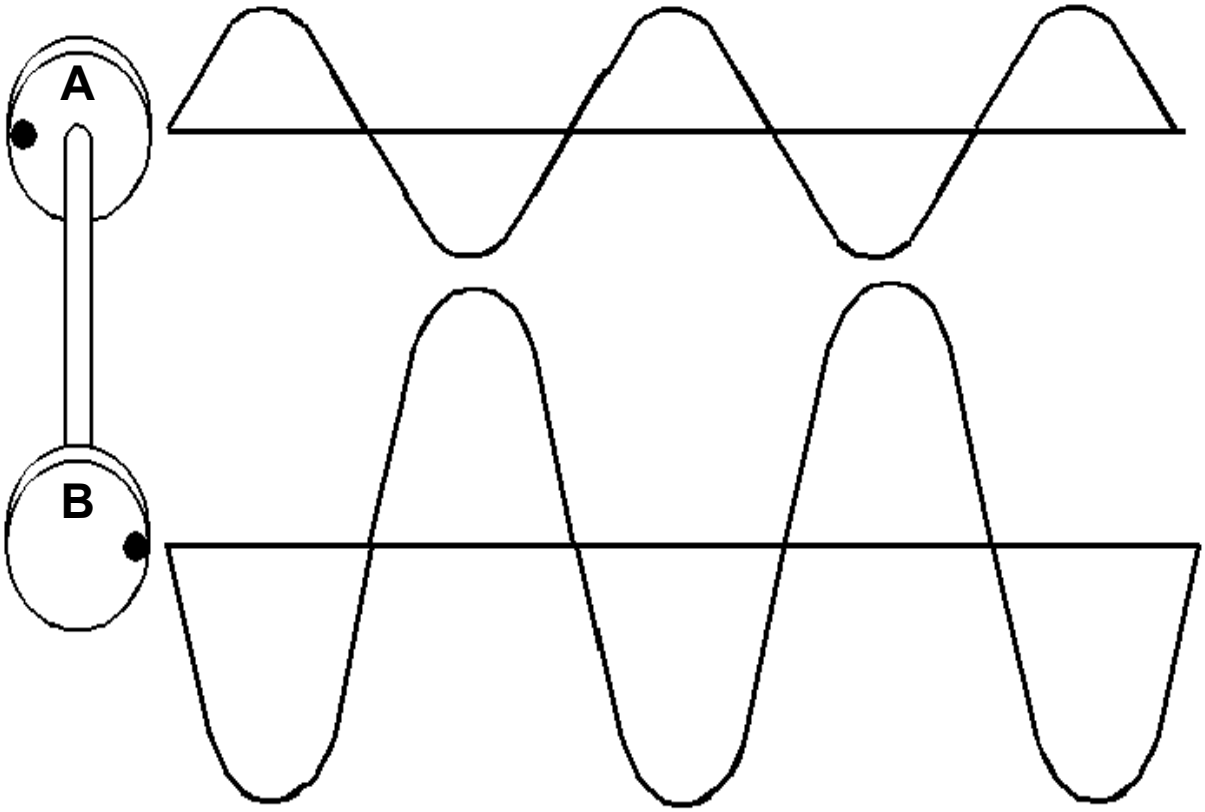


Figure 18

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Technical Components of Vibration Monitoring

Most vibration data collection systems acquire and trend the overall energy levels in rotating equipment. However, overall energy alone may not represent an accurate condition of the machine.

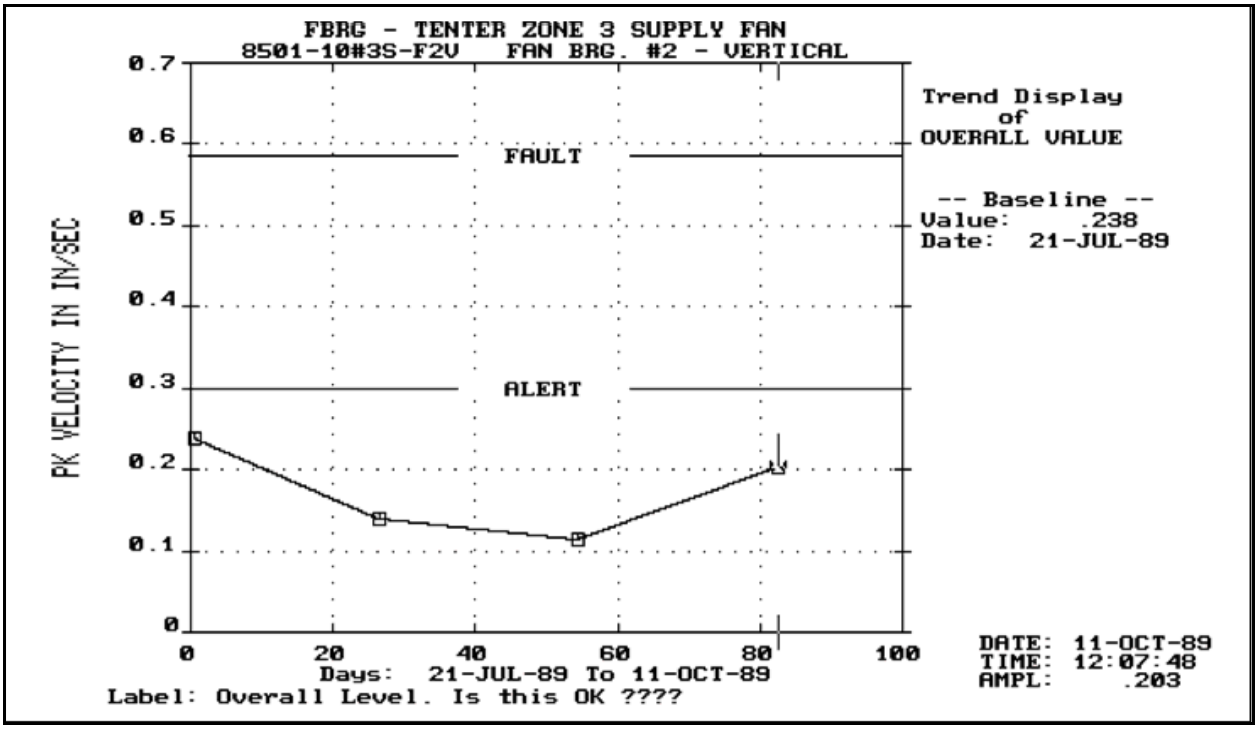


Figure 19

Based on the trend in Figure 19, determine the condition of this machine.

List some reasons for your assessment.

1.

2.

3.

4.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

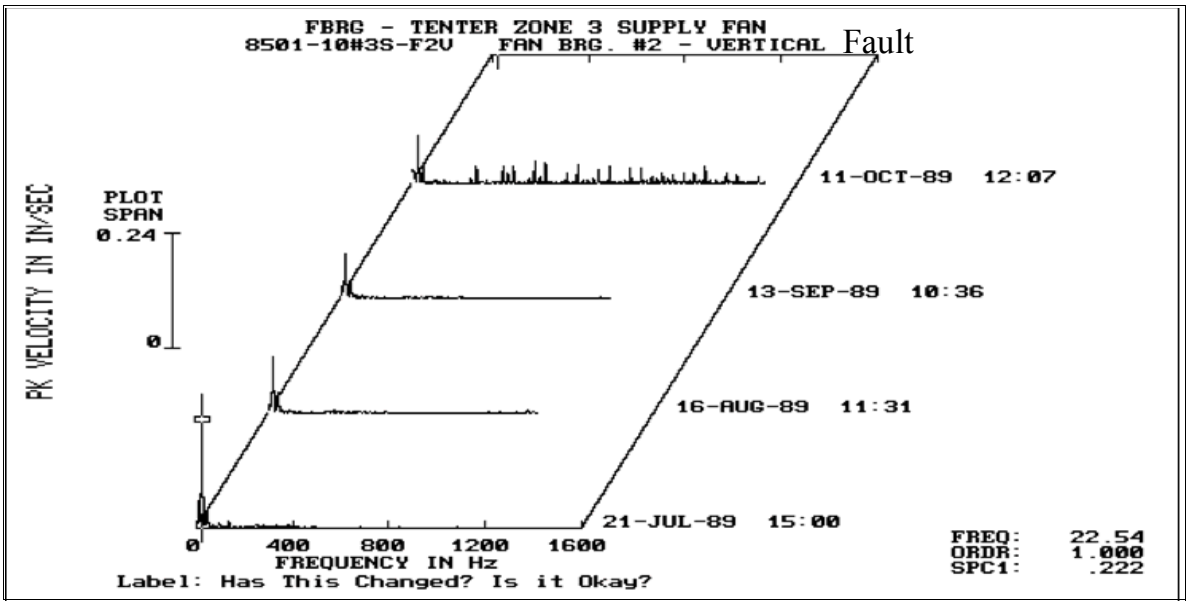


Figure 20

The ability to store and compare spectra greatly enhances any PDM program. For example, the spectra in Figure 20 represent the same data from the overall trend in Figure 19. The spectral comparison shows that, although the overall level decreased, the vibration characteristics have changed significantly. Note the increase in high frequencies and the decrease of the 1x turning speed (RPM or first order) peak. This evidence proves that neither the overall reading in Figure 19 nor that for 1x turning speed (TS) accurately assesses machinery condition. (See Figure 21)

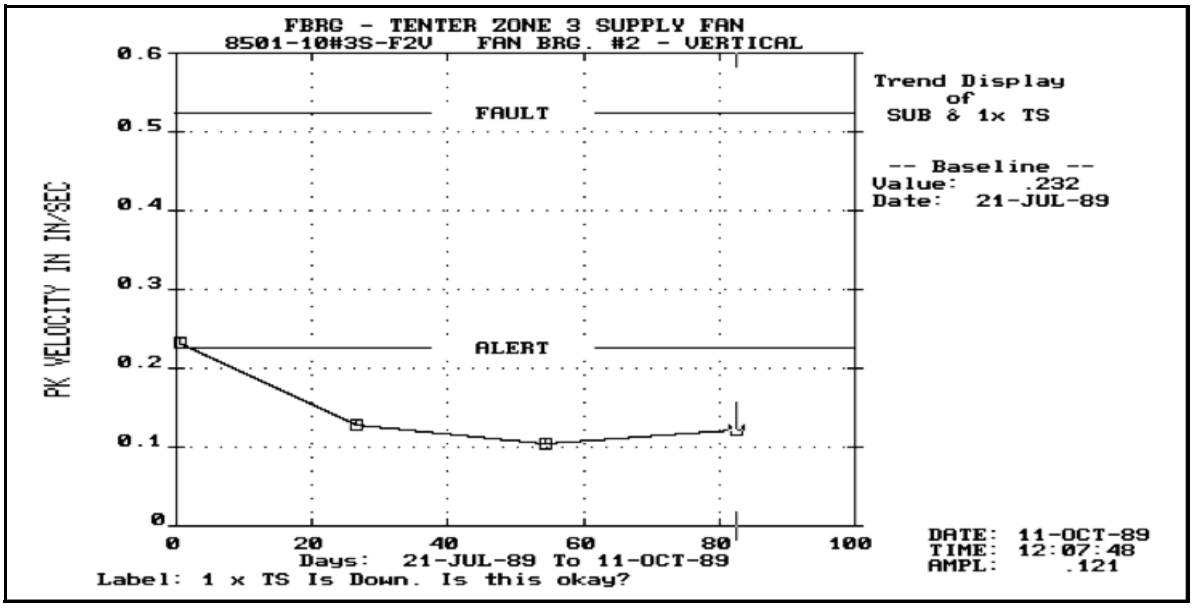


Figure 21

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

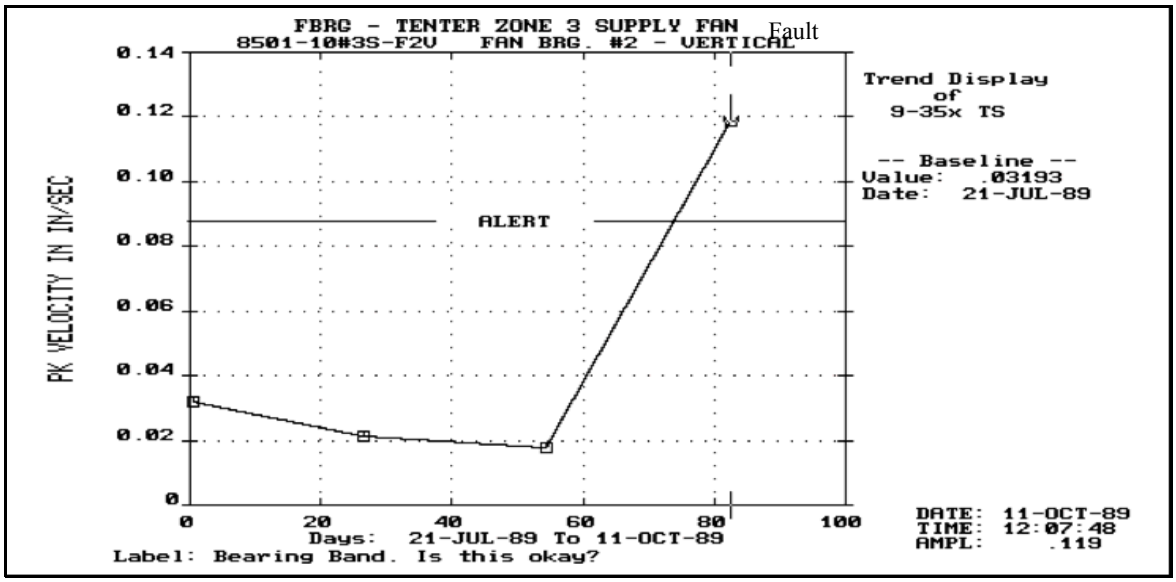


Figure 22

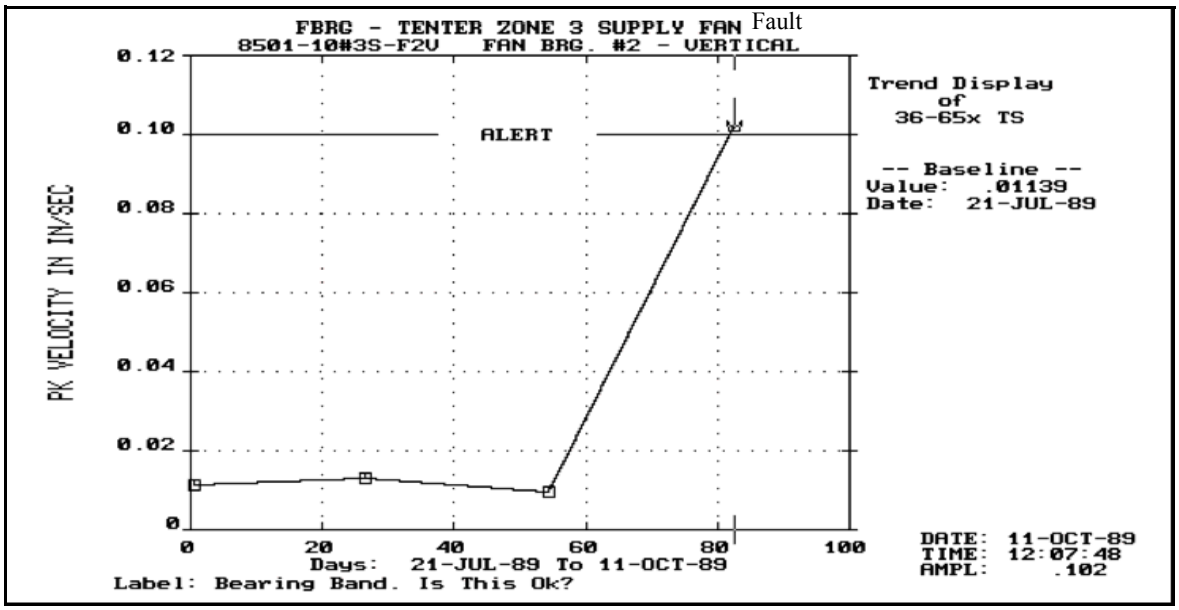


Figure 23

The ability to divide the overall value into selected frequency bands for more discrete alarming and analysis provides a powerful tool for vibration analysis. The trends in Figures 22 and 23 were defined for bearing detection. These alarms differ from those for the overall and for 1xTS shown in Figures 19 and 21. Again, these plots can be misleading without more complete data. This data should be a cause for alarm.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

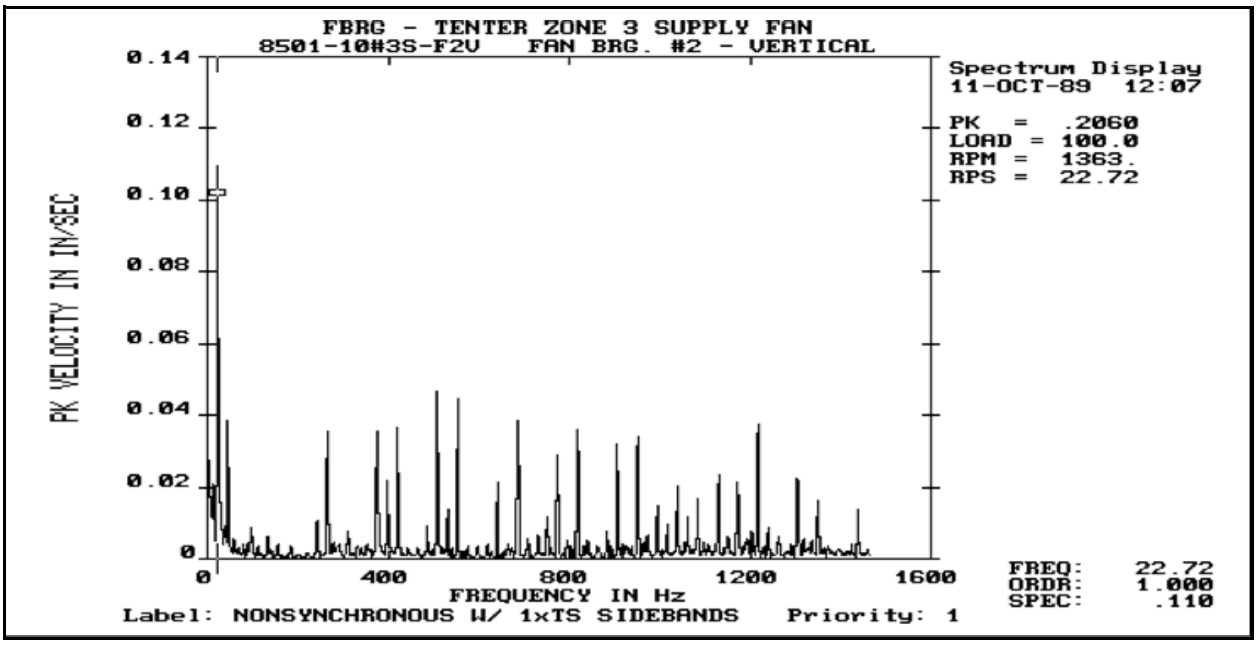


Figure 24

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	4.72	.0300	.21	13	685.97	.0408	30.19
2	9.71	.0170	.43	14	773.59	.0328	34.05
3	13.22	.0210	.58	15	818.68	.0368	36.03
4	22.71	.1094	1.00	16	906.28	.0320	39.89
5	45.06	.0435	1.98	17	951.30	.0373	41.87
6	265.35	.0357	11.68	18	1038.94	.0237	45.73
7	375.54	.0360	16.53	19	1084.01	.0171	47.71
8	397.98	.0242	17.52	20	1128.98	.0249	49.69
9	420.57	.0386	18.51	21	1172.04	.0217	51.58
10	508.21	.0520	22.37	22	1216.74	.0415	53.55
11	553.30	.0462	24.35	23	1304.34	.0257	57.41
12	640.91	.0213	28.21	24	1349.51	.0166	59.40

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.2060	.0410 / 4%	.1381 / 45%	.1473 / 51%

Note: Runspeed must be located before using the Peak-List.

Table 3

With some diagnostic experience, bearing defects can be recognized by their high-frequency peaks and the number of non-synchronous peaks with 1xTS sidebands. With this in mind, it is not necessary to know the bearing ID, the number of balls, or other such information about the bearing.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The time domain or waveform plot provides yet another helpful vibration analysis tool. Very high levels of impacting and ringing appear in the waveform in Figure 25. Each time the ball or roller passes over the race defect, the vibration energy increases. The energy then decreases as the roller or ball passes away from the damaged area.

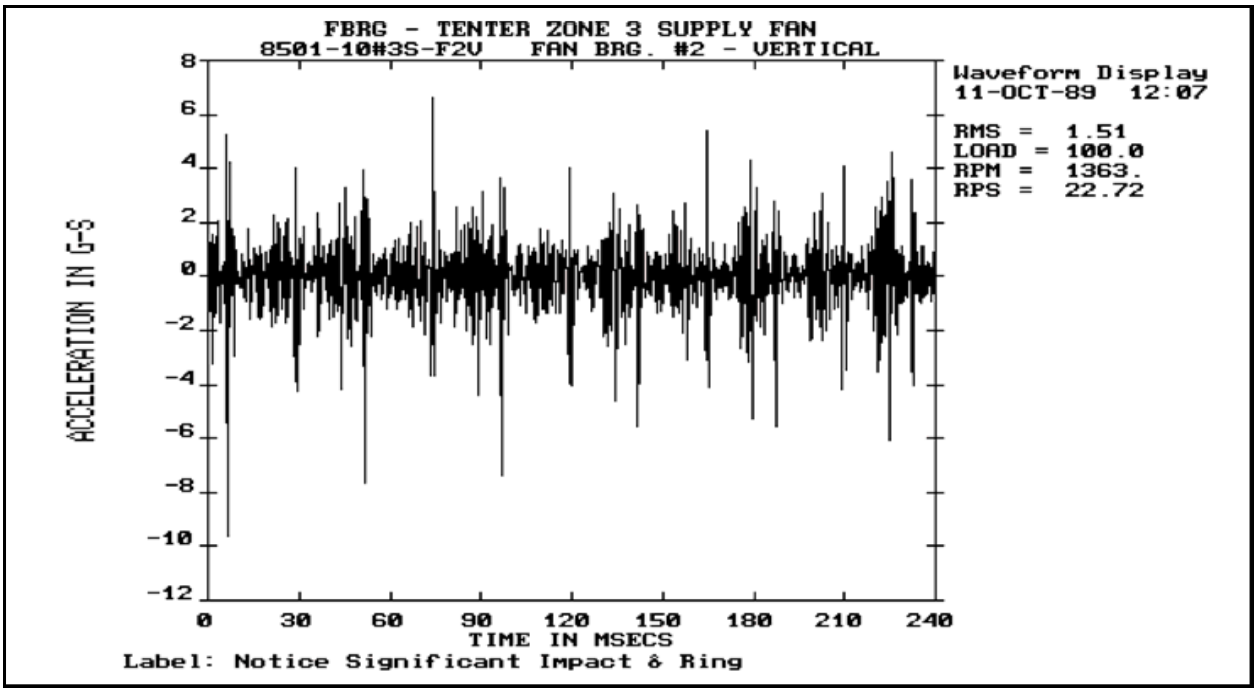


Figure 25

With some experience, this combination of evidence would cause some concern even though the overall trend level has decreased over the past four months.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

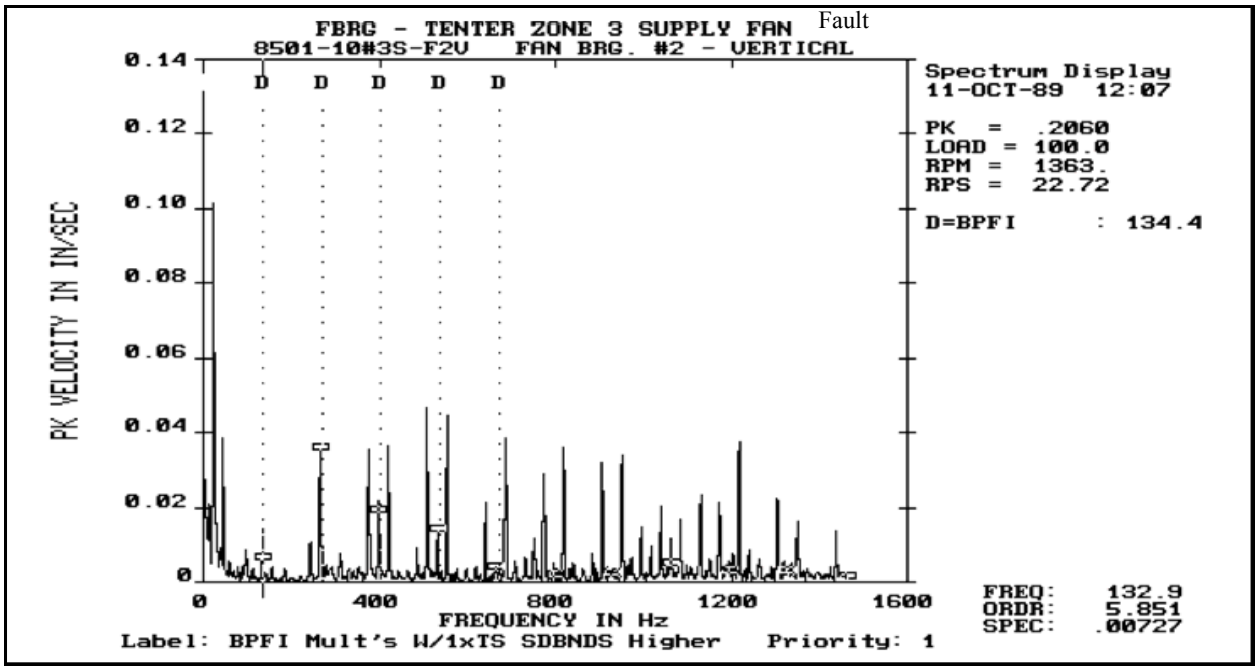


Figure 26

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	4.72	.0300	.21	13	685.97	.0408	30.19
2	9.71	.0170	.43	14	773.59	.0328	34.05
3	13.22	.0210	.58	15	818.68	.0368	36.03
4	22.71	.1094	1.00	16	906.28	.0320	39.89
5	45.06	.0435	1.98	17	951.30	.0373	41.87
6	265.35	.0357	11.68	18	1038.94	.0237	45.73
7	375.54	.0360	16.53	19	1084.01	.0171	47.71
8	397.98	.0242	17.52	20	1128.98	.0249	49.69
9	420.57	.0386	18.51	21	1172.04	.0217	51.58
10	508.21	.0520	22.37	22	1216.74	.0415	53.55
11	553.30	.0462	24.35	23	1304.34	.0257	57.41
12	640.91	.0213	28.21	24	1349.51	.0166	59.40

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.2060	.0410 / 4%	.1381 / 45%	.1473 / 51%

Table 4

An inner race defect can be accurately diagnosed when the bearing geometry associated with this bearing ID is known. Fault frequency overlays can also be used. Implementing the tools discussed in this course will help ensure the success of a predictive maintenance program.

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

An Effective Vibration Program

An efficient PDM program includes four major technical components.

- Consider the kind of transducer to use in each application. Choosing the proper transducer helps assure the collection of usable vibration data.

- Once the data is collected, the signal must be processed into a useful format. In most applications, the signal will be processed into either a time waveform or a spectrum for analysis.

- An important component of the PDM program involves problem detection. This component breaks down either the time waveform or the spectrum or both to determine whether a problem exists in the machinery.

- Whenever a problem is detected, utilize diagnostics. Diagnostics seek the source of the problem, the ***Root Cause of Failure***.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Review of Amplitude and Frequency Units

AMPLITUDE

- **Acceleration (G's)**
- **Velocity (Ips)**
- **Displacement (Mils)**

FREQUENCY

- **Cycles per Minute (CPM)**
- **Cycles per Second (Hz or CPS)**
- **Orders (Given Freq* / TS in RPM*)**

**To avoid confusion, the units for each variable should be the same. Hz and Hz or CPM and CPM.*

Frequency unit selection can be important. Occasionally viewing data in Orders vs. Hz or CPM makes analysis easier, depending on the defect. Analysts should be familiar with the way peaks are labeled and the how the cursor information is displayed in the data using the various units.

Changing amplitude units has a significant effect on the appearance of the data with respect to low frequency vs. high frequency peaks and amplitudes.

Note

The terms RPM and CPM are often used interchangeably. In some cases however, a CPM count will not equal an RPM count. For example, an automobile engine has a rotating frequency (RPM) but the pistons do not rotate; they reciprocate or travel in a linear fashion. Their frequency is referred to as CPM not RPM. There are other examples of this difference that will be covered later in the course.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Types of Transducers

Industry benefits from the availability of a number of transducers. The most common transducers read either displacement, acceleration, or velocity. Although these three types of transducers differ in their characteristics, every transducer works by converting mechanical energy into an electrical signal. Once it converts the signal, the transducer should render an accurate reading in its type of units.

Displacement Transducer / Prox Probe

A displacement transducer measures actual shaft movement relative to a transducer reference point. A sleeve bearing offers the best application for this non-contact probe. The advantages and disadvantages on page 1-28 must be weighed when considering the use of this transducer.

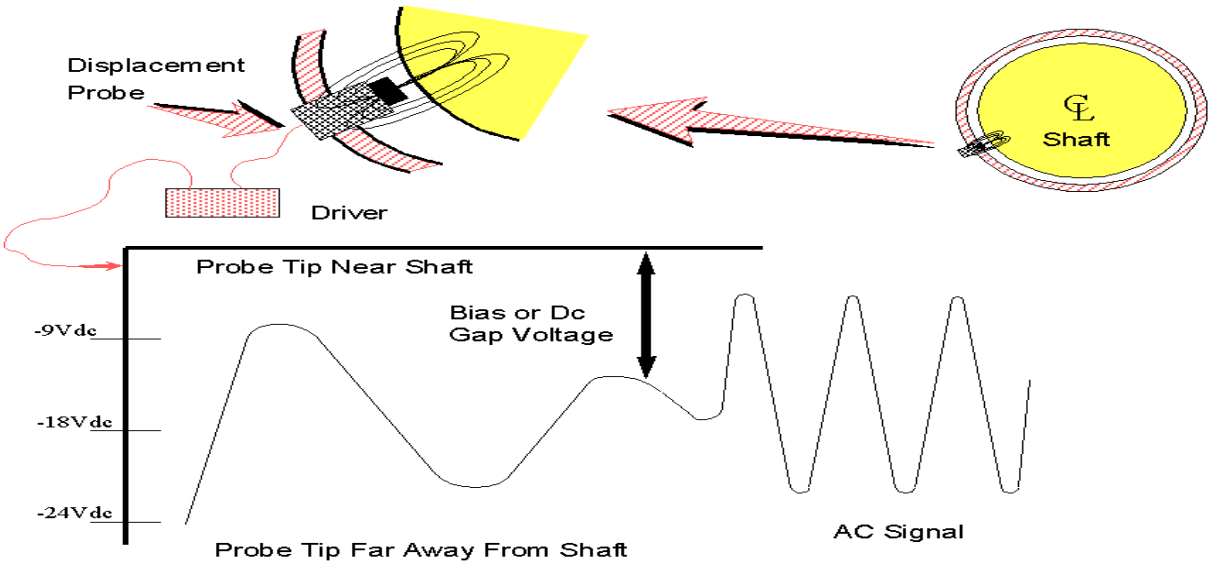


Figure 27

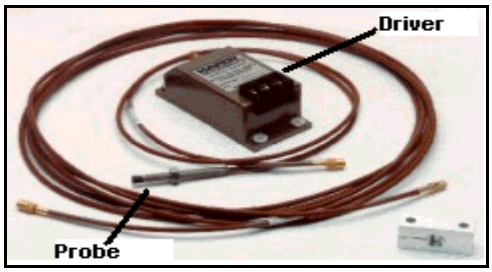


Figure 28

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The displacement probe with the electrical power provided to the probe tip generates a magnetic field. As it vibrates, the shaft passes through the magnetic field, which causes an electrical signal proportional to the vibration of the shaft. Typical sensitivity of a displacement probe is 200 millivolts per mil with a gap voltage within the middle of the power supply source. (See Figure 29)

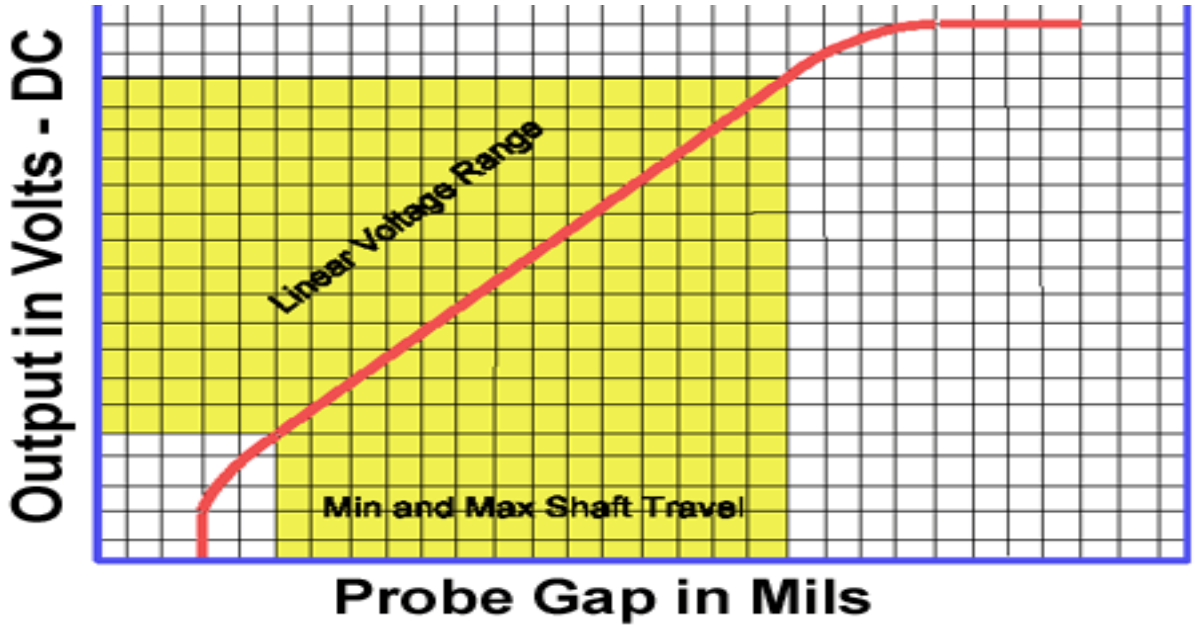


Figure 29

Figure 29 illustrates how to obtain the best linear response from the Displacement Transducer/Prox Probe.

Advantages

- Measures the relative motion between the probe tip and the rotating shaft; ideal for machinery with journal bearings.
- Extremely useful when little vibration transmits to the machinery case.

Disadvantages

- Requires permanent installation, which often proves difficult and sometimes impossible.
- The frequency response is typically linear from DC to 1,000 Hz.
- Requires an electrical source and signal conditioning affected by electrical runout.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Seismic Velocity Transducer

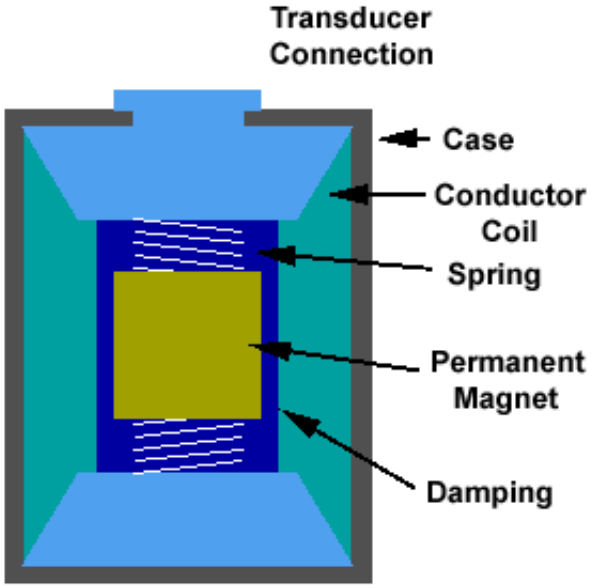


Figure 30

Use the velocity transducer when the actual shaft vibration cannot be observed. Use it also when sources other than the component shaft generate the vibration signals. Always consider the amount of energy being absorbed by the machine support or by the structure itself. The velocity of the machine case or bearing housing provides the key parameter. Velocity measures how fast the object or mass crosses the equilibrium (reference) point.

Like all other electromechanical devices, the velocity transducer has advantages and disadvantages. You must assess them accurately to determine the applications best suited to the velocity transducer.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Seismic Velocity Transducers

Advantages

- Among all transducer types, the signal-to-severity ratio is the closest to one-to-one
- Has an excellent signal-to-noise ratio
- Requires no external power supply
- Only single differentiation or integration required to go from velocity to another parameter type (Integration and Differentiation will be discussed later in this section.)
- Very rugged construction

Disadvantages

- Very large size
- Typically heavy
- The frequency range is limited to approximately 10 Hz to 2000 Hz, depending upon the type of transducer
- Excessive external temperatures affect the linear response of the transducer signal
- Relatively expensive compared to other transducer types
- An external magnetic field may affect the electrical signal
- The output signal may be altered by the orientation of the transducer; must be mounted horizontally to obtain the best results
- Wear and temperature fluctuations may cause frequent changes in the calibration

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Accelerometers

These transducers provide an electrical charge proportional to acceleration by stressing piezoelectric crystals. When a high force results in a small displacement or velocity (e.g., gears), acceleration gives the best measure of the force associated with the vibration. Basically, acceleration measures how fast an object comes to a stop at the peak of each cycle. Acceleration can be defined as how fast the vibrating component changes velocity in a given time frame.

Note

CSI analyzers can be configured to recognize a strobe light. At the appropriate command the strobe will flash at a selected frequency

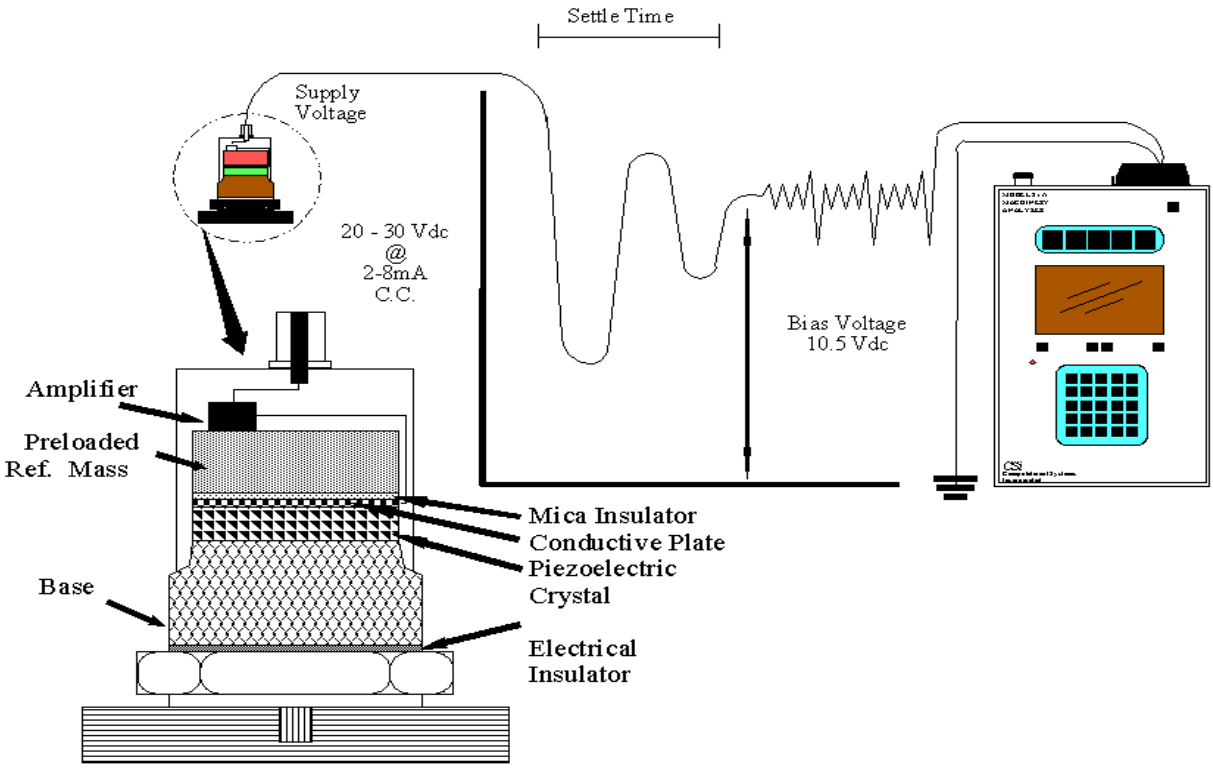


Figure 31

The vibration signal is sent from the accelerometer to the Analyzer as a voltage signal. The Voltage is divided by the sensor sensitivity then converted into the units defined at the measurement point set-up, either **Displacement, Velocity,** or **Acceleration.**

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Accelerometers

Advantages

- Possesses a broad frequency range from approximately 1 Hz to 30 kHz and higher depending upon the mounting technique used for the application. You should know the frequency ranges of the accelerometer you are using. There should be a transducer specification sheet that came with the transducer.
- Very rugged, small, lightweight
- No external signal conditioning required (Integrated Circuit Piezoelectric [ICP] type)
- Easily mounted with a stud or adhesives; magnetic mounts also available for periodic applications

Disadvantages

- Provides very poor signal response when used as a hand-held probe on high frequency components
- Limited signal-to-noise ratio
- Reads acceleration
- Requires double integration to cross all vibration parameters
- Requires an external power supply

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Accelerometer Mounting Response

Each accelerometer has a different response characteristic depending upon the mounting technique used for data collection. Figures 32 through 36 are spectral plots of actual accelerometer responses and the methods used for mounting each transducer.

Stud Mount

The spectral data in Figure 32 was produced with an accelerometer stud mounted on a smooth surface. It provided a linear response to approximately 16,000 Hz.

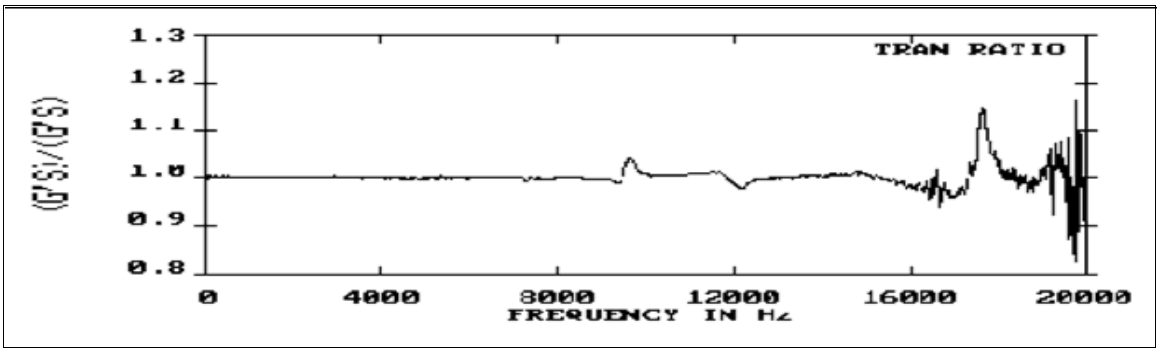


Figure 32

Quick Lock Mount

The spectral data in Figure 33 came from an accelerometer mounted with a CSI Model 910 and 911 Quick Lock. The linear response of the transducer was repeatable to approximately 10,000 Hz.

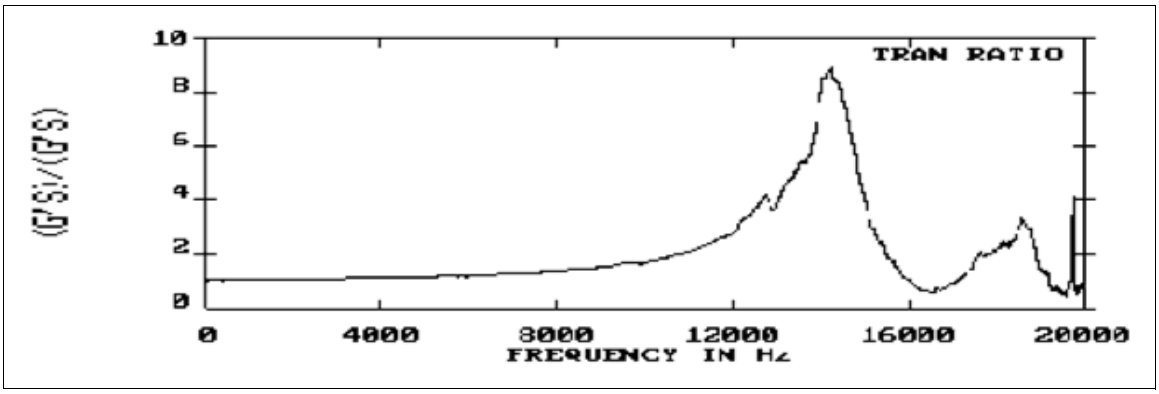


Figure 33

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Rare Earth Magnet Mount

The spectral data in Figure 34 was produced with an accelerometer mounted with a CSI Model 905 1 inch diameter Rare Earth Magnet. The linear response of the transducer went to approximately 7,000 Hz.

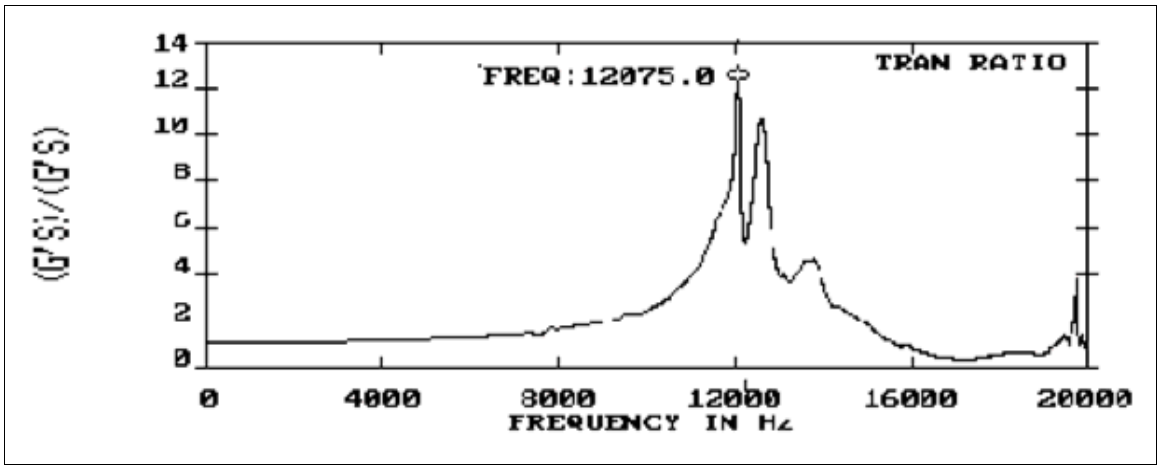


Figure 34

Super Magnet Mount

The spectral data in Figure 35 shows a linear response to approximately 3,000 Hz. The transducer was mounted with a CSI Model 906 Super Magnet on a curved surface. This is the large square 2 pole magnetic base.

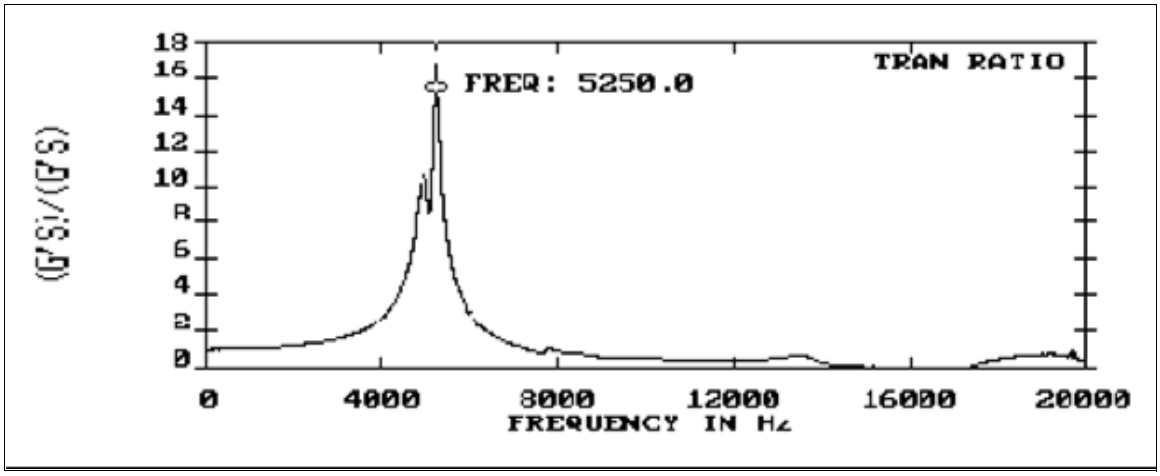


Figure 35

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Hand-Held Accelerometer with 2" Stinger

The data in Figure 36 came from a CSI Model 310 hand-held accelerometer using a 2" steel stinger. The response of the transducer is linear to approximately 800 Hz. For high-speed equipment, this F_{max} is not acceptable. Also the model 310 is difficult to hold with the same amount of pressure and hold it perpendicular to the shaft each time you collect data. Only use the Model 310 if it is the only means of collecting data.

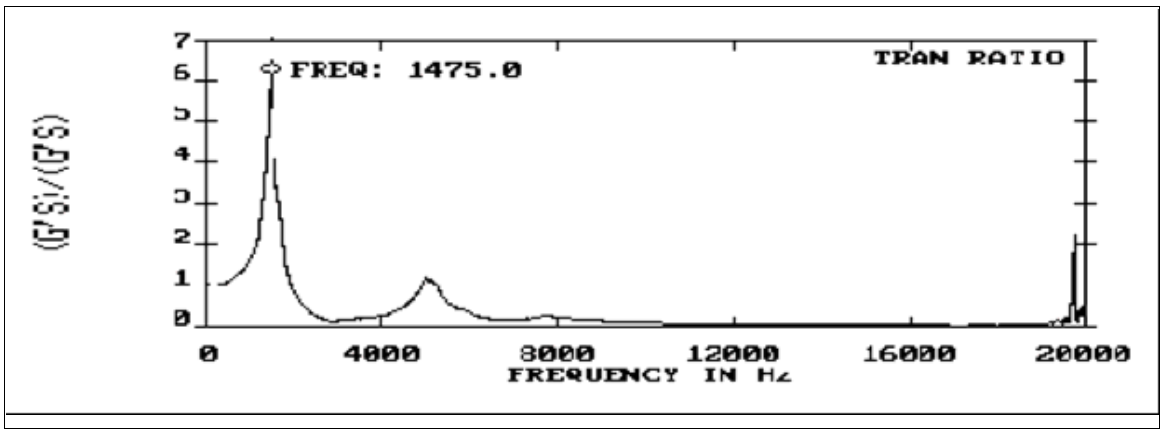


Figure 36

Hand-Held Accelerometer with 8.5" Stinger

The data in Figure 37 came from a CSI Model 310 hand-held accelerometer using an 8.5" steel stinger. The response of the transducer is linear to approximately 500 Hz. For high-speed equipment, the F_{max} of the hand-held probe is not acceptable. It is even more difficult than the 2 inch stinger to hold with the same amount of pressure and hold it perpendicular to the shaft each time you collect data.

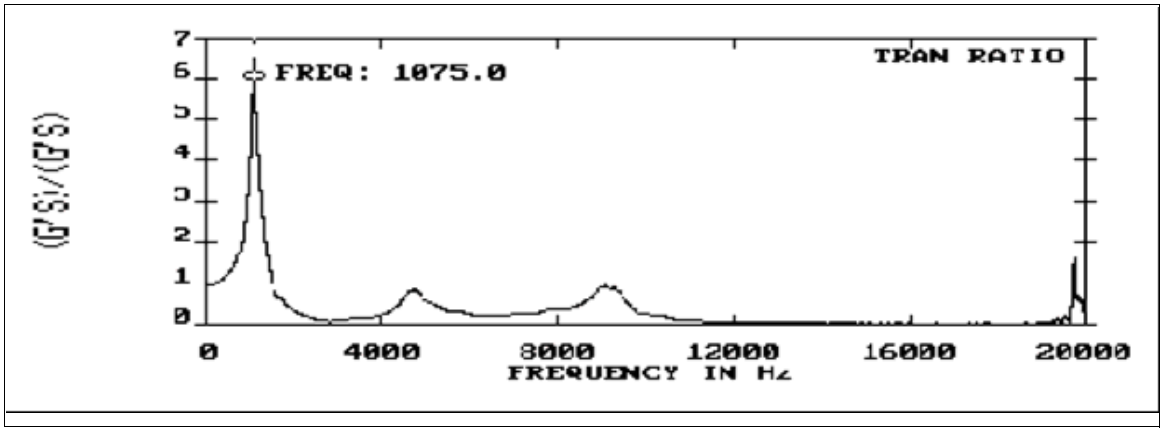


Figure 37

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Signal Processing

All forms of signal processing perform the same function - translate the transducer output signal into a more understandable format. The four primary types of processed signals for vibration analysis include:

- **Time domain display (waveform)**
- **Overall level criteria**
- **Selective frequency band analysis**
- **Frequency domain display (spectral analysis)**

Time Domain Signal

The time waveform in Figure 38 measures the amplitude of a voltage signal over a period of time.

The voltage is divided by the sensitivity to obtain the amplitude in the sensor units.

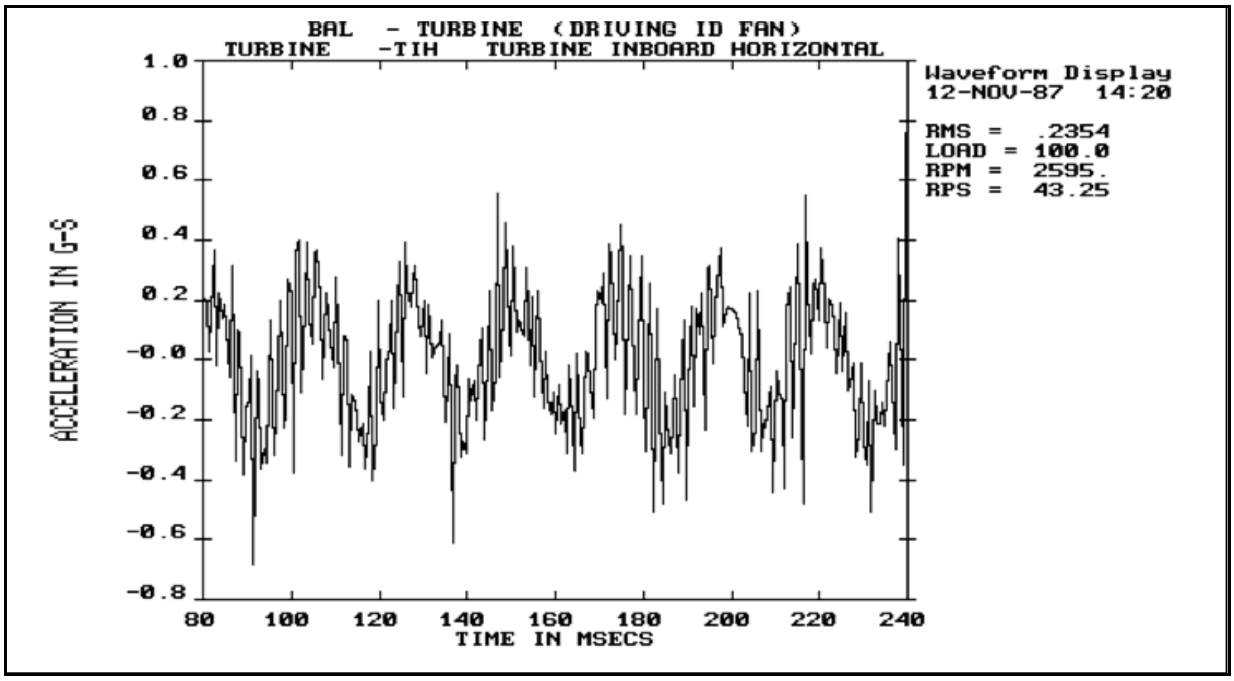


Figure 38

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The time domain signal gives important data. For example, the waveform in Figure 39 indicates there may be a bearing defect because of its high G level of impacting.

Impacting levels with an amplitude swing of approximately 2 G's are usually cause for concern on a pump or a motor. Gearboxes, however, tend to generate much higher G levels because of the constant meshing (tooth contact) of the gear teeth.

In Figure 39, there is an approximate G swing of 16 g’s. Rolling element bearing defects commonly generate similar patterns.

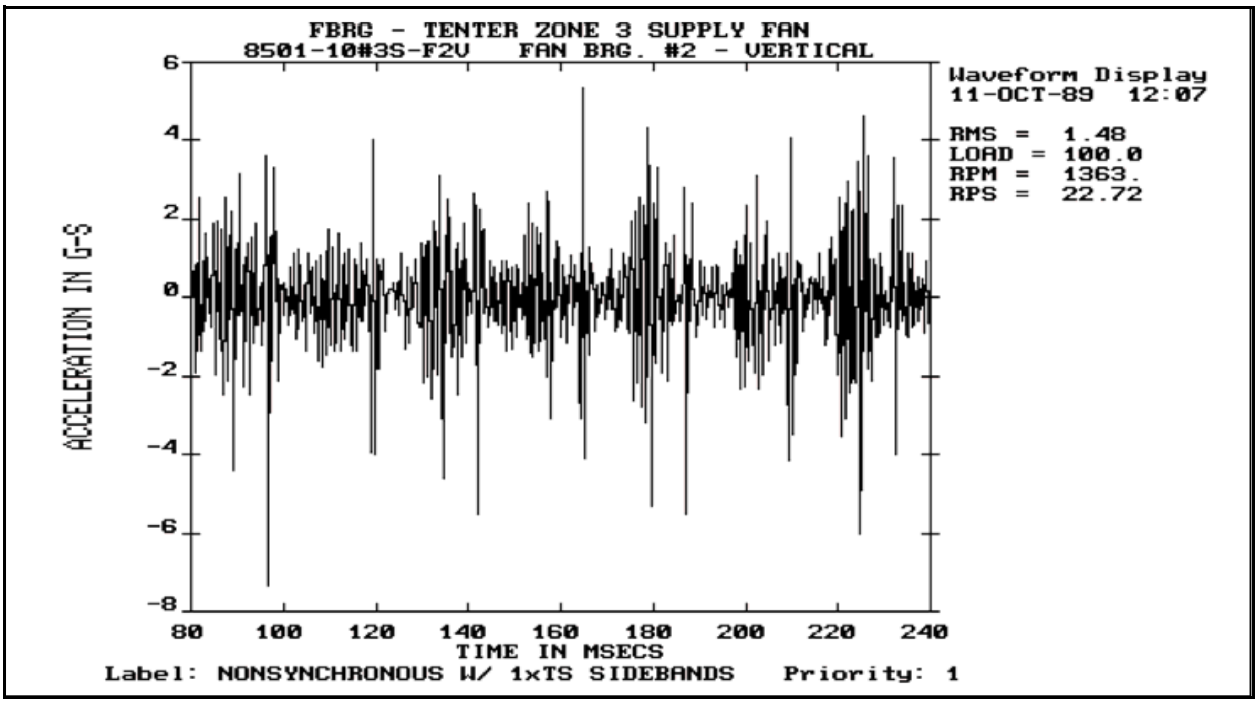


Figure 39

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Overall Level Criteria

The overall level is a single number calculation of the unfiltered amplitude of a vibration waveform. The overall level of a spectrum can also be calculated. Several organizations have used overall level criteria to establish many different standards for machinery levels.

Caution!
Be very careful when assigning alarm values to your equipment. Similar machines can run at different levels (amplitude) of vibration.

Summary of Overall Vibration Standards

Velocity (in/sec) Peak

Standard	Measurement	Alert Level	Alarm Level
Hydraulic Inst. 14th Edition	Casing	0.30*	----------
I.S.O. 2372	Casing	0.25	0.60
E.P.R.I. FP 754	Shaft	0.50	0.80
A.P.I. 610 6th Edition	Shaft	0.40	----------
Rathbone Chart	Casing	0.30	0.60
*Filtered reading valid 2,000 - 20,000 CPM			

Table 5

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

A PDM program is only as good as the standard and equipment upon which it is based. An insurance agent responsible for insuring companies and their equipment established the chart in Figure 40 in 1939. So he could set an adequate premium, he had to know the running condition of the machinery. The agent based his chart on casing measurements made on heavy, slow-speed machines. The chart was fine for its intended purpose, but it is inadequate for a wide range of machinery built for industrial purposes today.

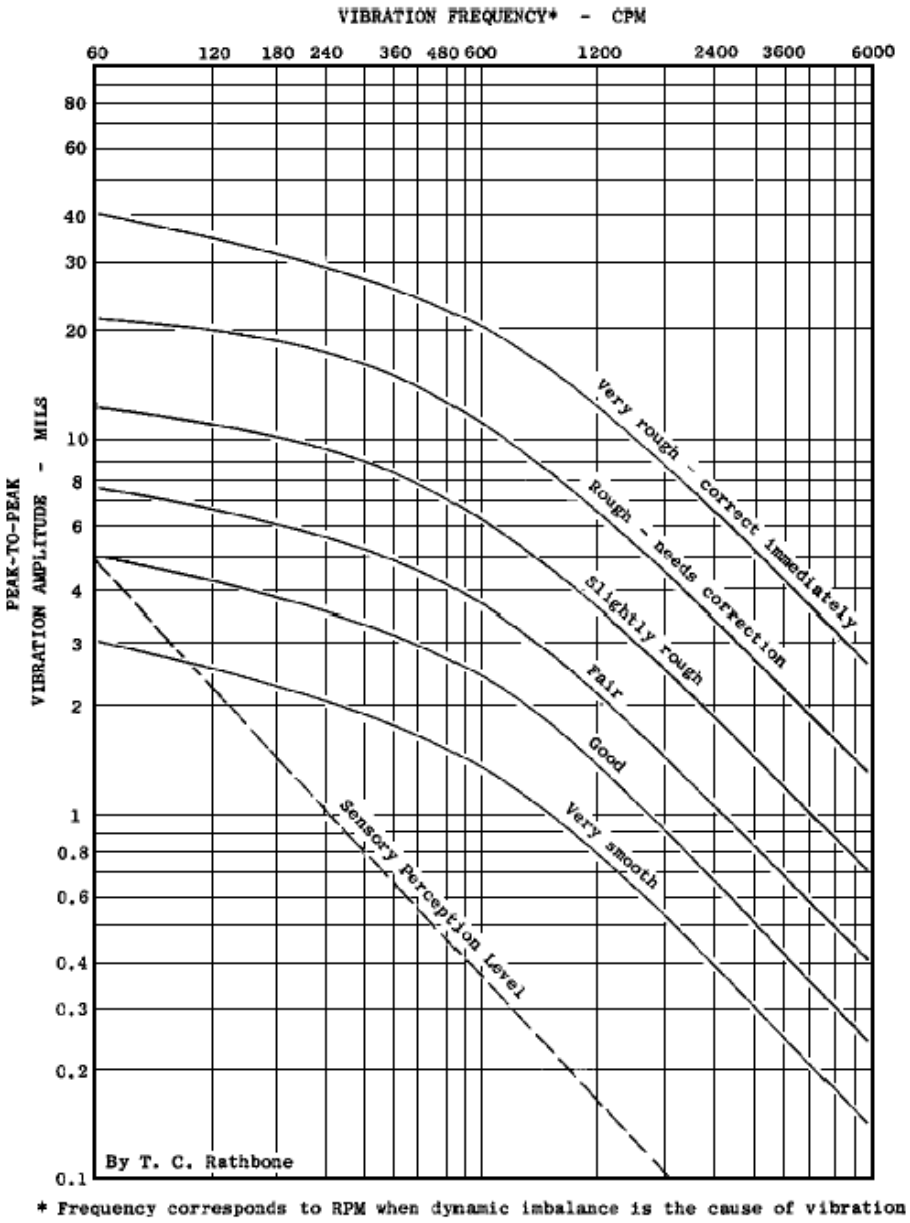
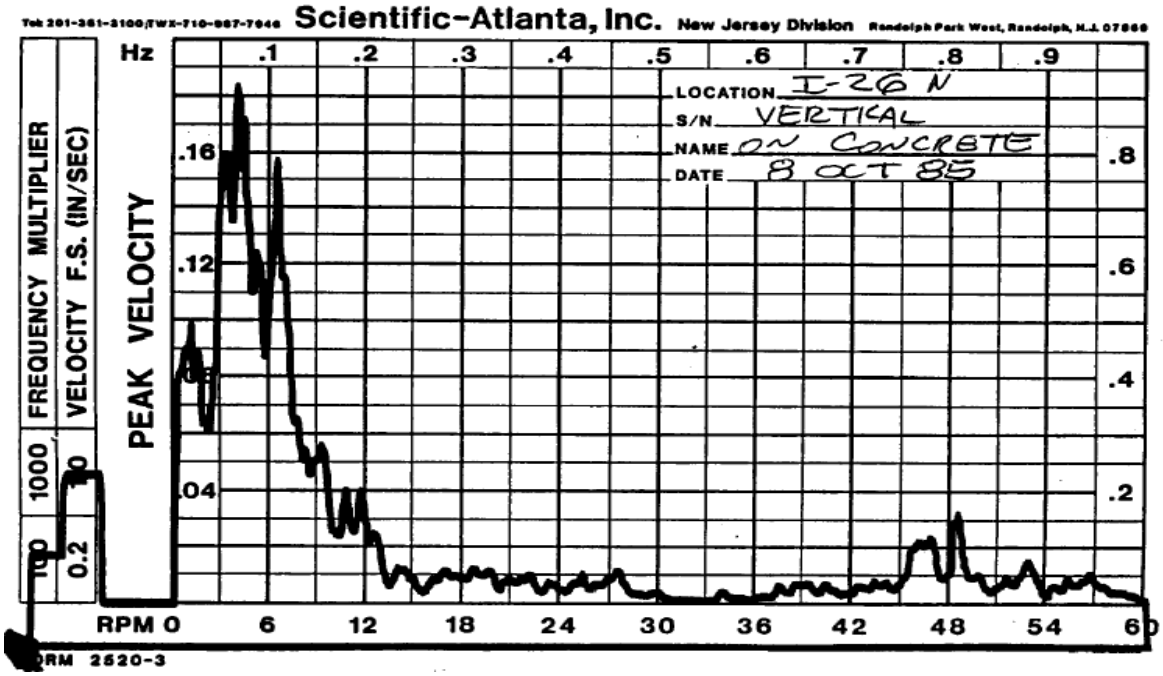


Figure 40

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Selective Frequency Band Analyzer

Some PDM programs use a selective frequency band analyzer (Swept Filter). The spectral data shown in Figure 41 was made using a swept filter analyzer. The broad peaks result from sweeping one filter through the entire frequency range of interest. The disadvantage is that the resolution in plot allows very poor analysis capabilities.



Frequency

Figure 41

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Frequency Signature/ Categories of Energy

A frequency domain signal is plotted with the vertical (Y) axis as the amplitude and the horizontal (X) axis as the frequency signature. The data contained in the frequency domain is derived from the time waveform. The frequency domain can be divided into three major areas of interest. (See Table 6

Synchronous Components	N x RPM (n is an integer)
Sub synchronous Components	< 1 x RPM
Non-synchronous Components	F x RPM (F > 1.0 but not an integer)

Table 6

Note
RPM (also called turning speed) is the rotating frequency of the shaft at the measurement point where you collect data.

Some Causes for Sub synchronous Components

These frequencies occur below 1 x RPM of the rotating shaft. Possible causes for sub-synchronous components include:

- Another machine
- Another component in the monitored machine
- Machines with belts have a primary belt frequency and, often, a 2 x belt frequency
- Hydraulic instability, such as oil whirl and oil whip
- Rotor rub, shaft rub, compressor wheel rub
- The cage frequency of antifriction bearings

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Some Causes for Synchronous Components

These frequencies are integer multiples of the running speed of the machine. These defects are always exact multiples of RPM (N x RPM where N is an integer).

Lower multiples - n = 1 to 8

- Imbalance
- Pitch line runout
- Misalignment
- Bent shaft
- Looseness
- Blade or vane pass
- Reciprocating motion

Higher multiples - n > 8

- Gears
- Blade pass
- Slot frequency of motors

Some Causes for Non synchronous Components

These frequencies occur above the run speed of the machine, but they are not integers of running speed (F x RPM, where F > RPM but not an integer).

- A component on another machine
- Multiples of belt frequency
- Antifriction bearings (#1 defect you will find on equipment)
- System resonances
- Electrical
- Chain drives
- U-joints
- Centrifugal clutches
- Lube pumps
- Compressor surge
- Detonation
- Sliding surfaces

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Harmonics and Orders

Harmonics are frequencies that occur at integer multiples of some fundamental frequency (1 x F, 2 x F, 3 x F, etc.).

Harmonics: f = N x f

Where f is a given frequency, and N is some integer (1, 2, 3, 4, etc.)

Figure 42 illustrates data displayed as frequency in orders. The labeled peaks are *synchronous* in nature, in that they are integer multiples of shaft speed. This integer multiplier also qualifies them as harmonics.

Harmonics vs. Orders

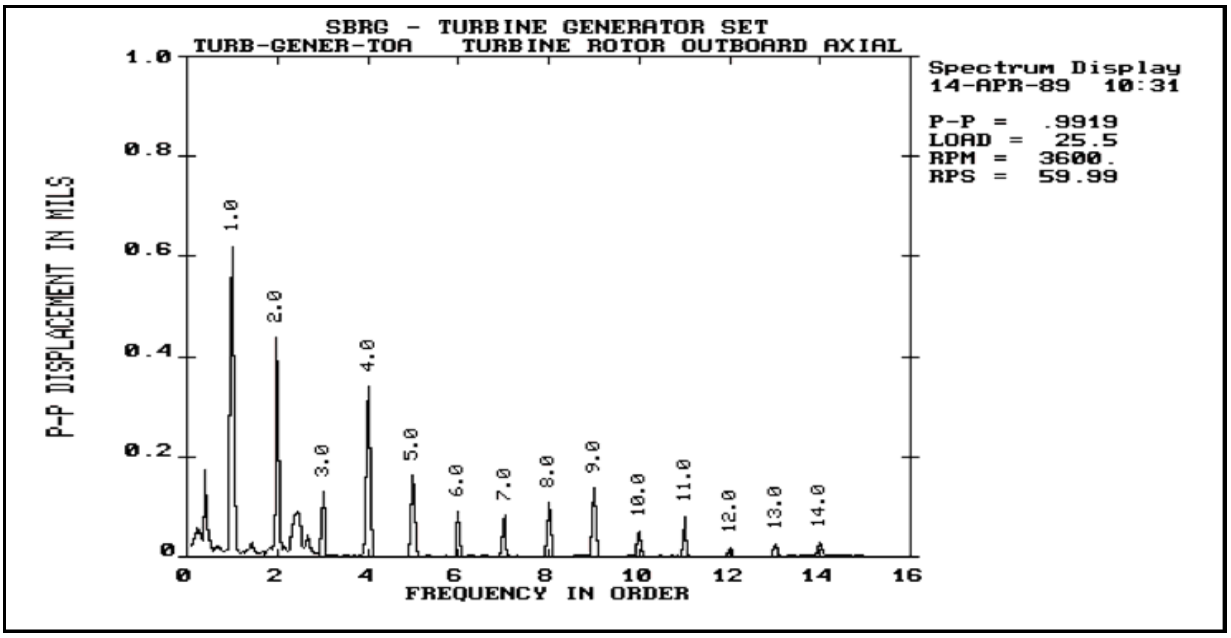


Figure 42

Orders are multiples, not necessarily integers, of turning speed of the shaft being monitored (1 x RPM, 2 x RPM, 3 x RPM, 4.56 x RPM, 33.68 x RPM, etc.). These values may be expressed as 1 order, 2 orders, 3 orders, 4.56 orders, 33.68 orders, etc. Orders are any frequency's relationship to turning speed. Figure 42 illustrates spectral data displayed as frequency in orders with synchronous peaks labeled. The labeling shows their relationship to shaft turning speed.

Orders: TS x any number

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The spectral data in Figure 43 displays non synchronous harmonics.

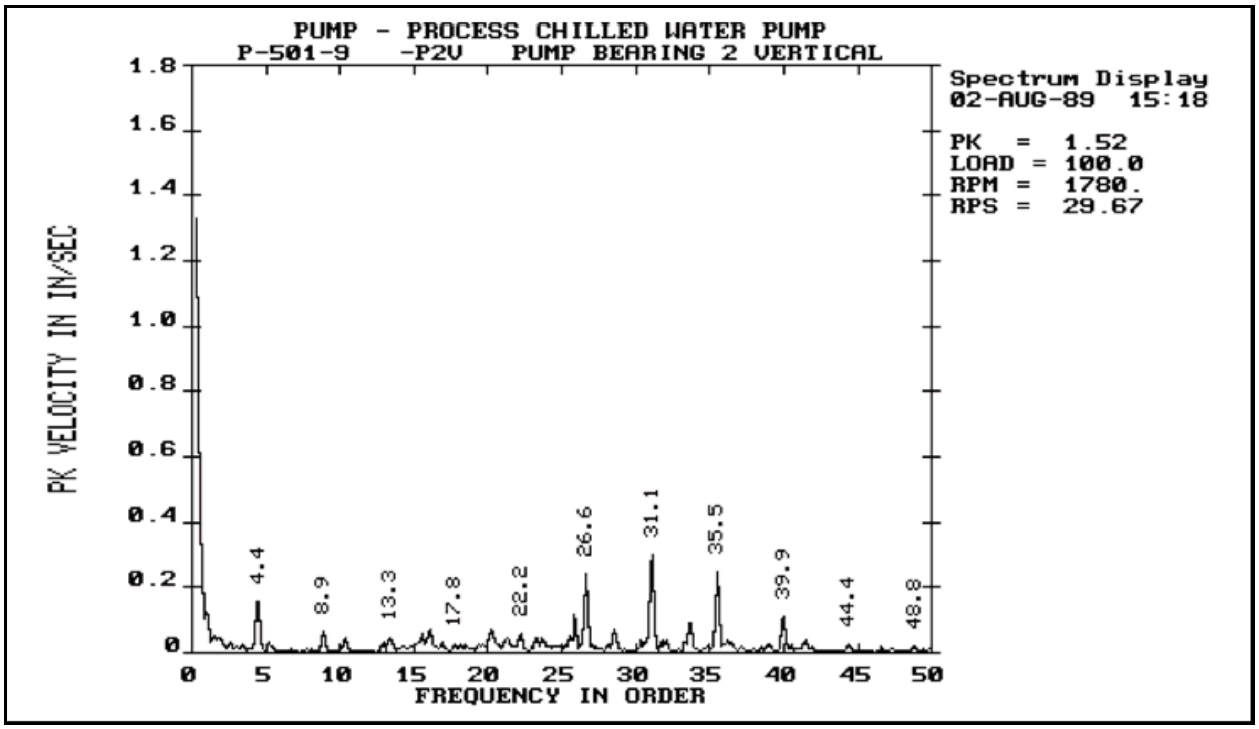


Figure 43

All labeled peaks are harmonics of the primary frequency 4.4 orders.

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Problem Detection

Without sufficient data, confirming a defect can be a challenge. However, it need not be difficult. Often, a simple diagram of the equipment can be a great help in diagnosing machinery problems. A machine diagram should include information such as:

- Estimated rotor weights
- Shaft diameters
- Bearing details
  - wType (sleeve or rolling element)
  - w Size
  - w Lubrication
- Operating frequencies
- Motor information
  - w Number of stator slots
  - w Number of rotor bars
  - w Slip frequency
- Turbine blade/ bucket count
- Belt / chain data
  - w Shaft center to center distance
  - w Pitch diameters
  - w Number of belts
- Coupling information
- Gear data
  - wTrain layout
  - wTypes of gears
  - wGear tooth count

Reminders for Data Collection

For an effective PDM program, data must be collected in the correct plane and in a consistent manner. Some faults show the highest amplitudes in the radial directions while others show up in the axial direction. If possible, collect two radial readings per bearing and one axial reading per shaft for each machine component in the train.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Walk Around Inspection

During data collection take note of the condition of the machine. Some things to look for are:

- General care and condition of the machine
- Structural integrity
  - Foundation
  - Cracked grout
  - Mounts and fasteners
- Leaks - lubrication, product, etc.
- Instrumentation - pressure, vacuum, flow, temperature

Operators can be a good source of valuable information. Frequently they will have a record of a history of the machine. Associating vibration signatures with this data can help resolve some problems.

- The last thing done to the machine
- History of the machine - recent changes in behavior
- Bearing clearances
- Lubrication practices
- Recent repairs
  - Shaft
  - Gears
  - Coupling
  - Belts
  - Alignment - How and why?
  - Vibration related

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Problem Confirmation

After compiling all pertinent information, follow a logical process to reach a viable conclusion. Ask yourself:

1) Is the problem real?
2) What is the problem?
3) How bad is the problem?
4) When should the problem be corrected?

Corrective Action

Mechanical defects such as imbalance, misalignment, looseness, and bearings generate a reasonable well defined vibratory pattern. It is common for machinery to suffer from multiple faults. When possible, find and repair one defect at a time. Start with the most severe, or those with a higher priority first.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Transducer Location

Transducer location is critical. Ideally the transducer should be as close to the source of energy as possible. The “path” that the energy must travel to reach the probe is called the “transmission path”. Place the probe so that the path is as short as possible. Surfaces like thin sheet metal, bearing covers, and motor housings do not provide a good transmission path.

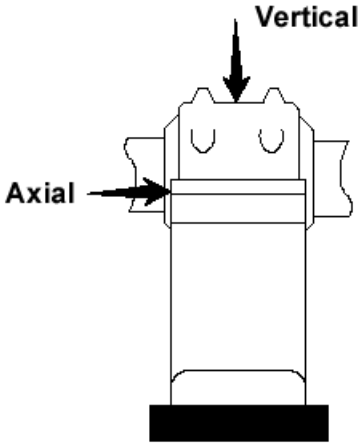


Figure 44

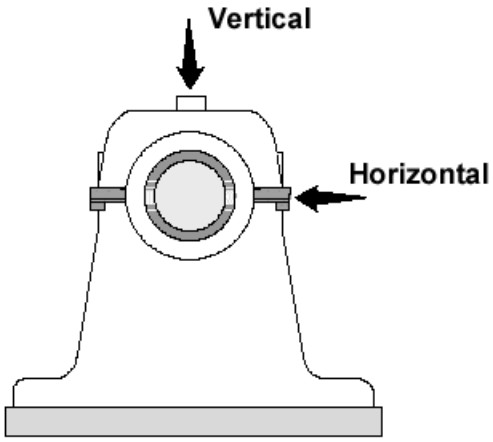


Figure 45

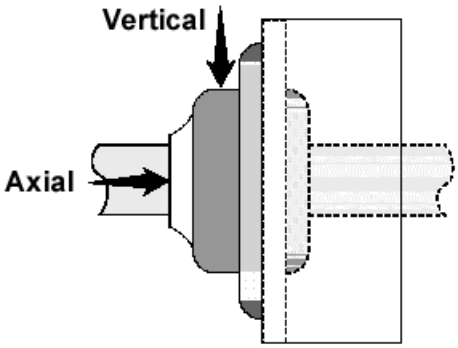


Figure 46

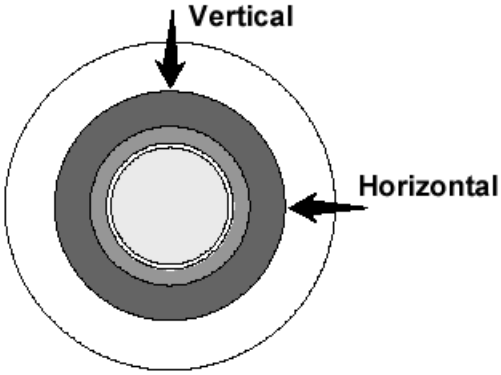


Figure 47

The small arrows in the figures 44 through 47 indicate measurement points for data collection. In general, always collect data in the three directions shown. The different orientations will help later in the diagnostic process.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Machine Data Sheet

Machine Data Sheet
page 1

Station: _____ Company Machine ID: _____

Machine Description: _____

Driver Type: _____ Mfr.: _____

Mfr. ID: _____ Horsepower: _____ RPM: _____

Comments: _____

Coupling Type: _____ Mfr.: _____

Comments: _____

Driven Equip. Description: _____
Mfr.: _____ Mfr. ID: _____

Service: Standby [ ] Intermittent [ ] Continuous [ ]

No. Vanes / Blades: _____

Comments: _____

Belt Type: _____ No. of Belts: _____

Gear Mfr.: _____ Mfr. ID: _____

Input / Output Ratio: _____ Input Speed: _____

Output Speed: _____

Comments: _____

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Machine Data Sheet page 2

Sketches

Please note gear information as applicable.

MPT	Point Description	Bearing ID or No. Teeth	Speed

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Fault Guide

Vibration	Dominant Frequency	Dominant Plane	Phase Reading
Unbalance			
Static	1xTS	Radial	Radial in phase
Dynamic	1xTS	Radial	Radial 0-180 out / 2 plane
Couple	1xTS	Radial/Axial	Radial 180 out
Overhung rotor	1xTS	Radial/Axial	Radial unsteady / Axial in phase
Misalignment			
Angular	1x, 2xTS	Axial	Axial 180 out
Offset	1x, 2x, 3xTS	Radial	Radial 180 out
Offset + Angular	1x, 2xTS	Radial / Axial	Radial / Axial 180 out
Sleeve Bearing	1x, 2xTS	Radial / Axial	Axial 180 out
Antifriction Bearing	1x, 2x, 3xTS	Axial	Axial 180 out
Bent Shaft	1x, 2xTS if on coupling end	Axial	Axial 180 out
Mechanical Looseness			
Non-rotating bearings	1 - 10 x TS	Radial	Radial
Rotating impellers	1 x TS predominant, as high as 10 x TS		
Antifriction Bearings	*Early stages* - Bearing frequency *Late stages* - 1 x TS and harmonics	Radial Axial on thrust bearing	
Sleeve Bearings			
Looseness	Multiples of TS	Radial	
Oil Whirl	0.43 x TS	Radial	
Belt Drives			
Mismatched, worn	2 x belt frequency	Radial inline with belt	
Eccentric sheave	1 x shaft speed	Radial	
Misalignment	1 x TS	Axial	
Gears - (GMF = Gear MEsh Frequency, SG = Spur Gears, HE = Helical Gears)			
Transmission error	GMF 1 + harmonics	Radial SG / axial HE	
Pitch line run-out	GMF + sidebands	Radial SG / axial HE	
Unbalance	1 x TS	Radial SG / axial HE	
Misalignment	1x, 2x TS	Radial SG / axial HE	
Faulty tooth	GMF + sidebands	Radial SG / axial HE	
Rotor Rub	0.5xTS and 1/2 multiples	Radial	
Electrical			
Loose iron	2 x line frequency (LF)	Radial	**Note:** There are several Electrical defects that appear at 2x LF.
Stator problems	2 x LF	Radial	
Phase unbalance	2 x LF	Radial	
Loose stator	2 x LF	Radial	
Broken rotor bar	2 x LF at 1xTS with sidebands	Radial	
Eccentric rotor	2 x LF at 1xTS with slipbands	Radial	
Loose slot	2 x LF, slot frequency + sidebands	Radial	
Pole pass	At 1xTS with sideband spacing = to # of poles x slip frequency		
Blade/Vane Pass	# of blades/vanes x TS	Radial	

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Unbalance

Section 2

Objectives

- Define Unbalance.
- Determine causes of Unbalance
- Identify spectral and waveform characteristics of Unbalance

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Unbalance

Unbalance occurs when the center of mass differs from the center of rotation.

Unbalance defined is the condition of a rotating component where the weight is unevenly distributed from the center of gravity. The center of rotation is not the same as the center of mass.

Some common causes of unbalance in rotating equipment are:

- Material buildup
- Wear
- Broken or missing parts
- Improper assembly
- Thermal distortions

Characteristics of Unbalance:

- Directional in nature, usually horizontal
- Turning speed peak amplitude changes with speed
- Little axial energy with center hung machines

Waveform:

- Simple, sinusoidal, periodic
- One event per shaft revolution
- Little to no impacting

Spectrum

- Elevated turning speed (1 x TS) peak amplitude
- Little to no turning speed harmonics

Note
Suspect other or additional defects with the presence of significant turning speed harmonics.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Case History #1 - Motor Driving Blower

The multiple point spectral plot in Figure 1 shows data from one inboard motor point and the blower measurement points. The dominant peak is related to turning speed (1 order). The strongest vibration occurs in the horizontal plane throughout the machine.

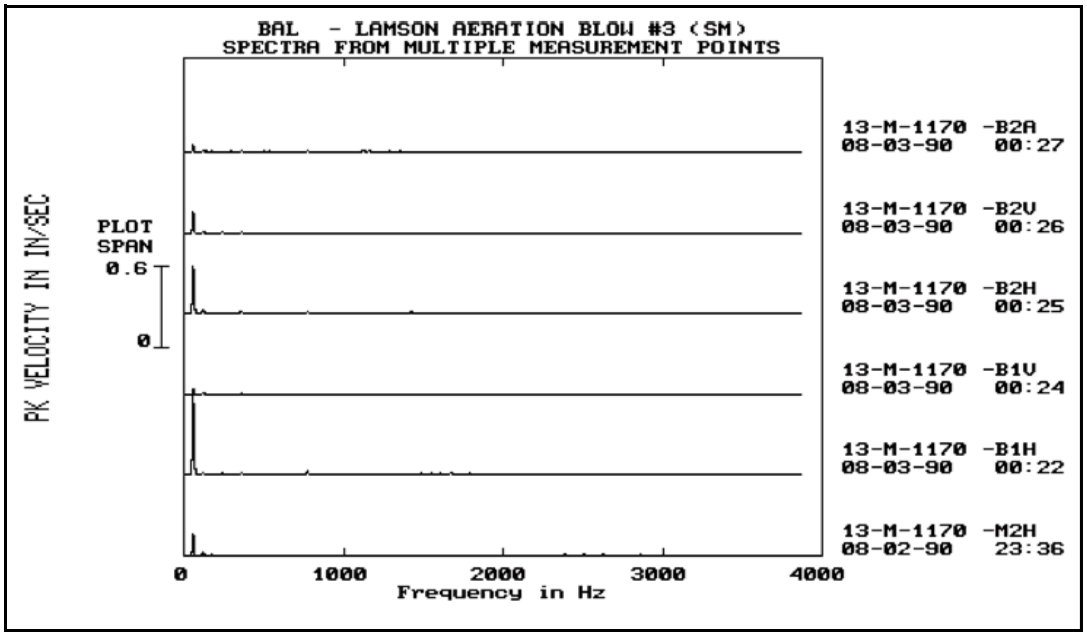


Figure 1

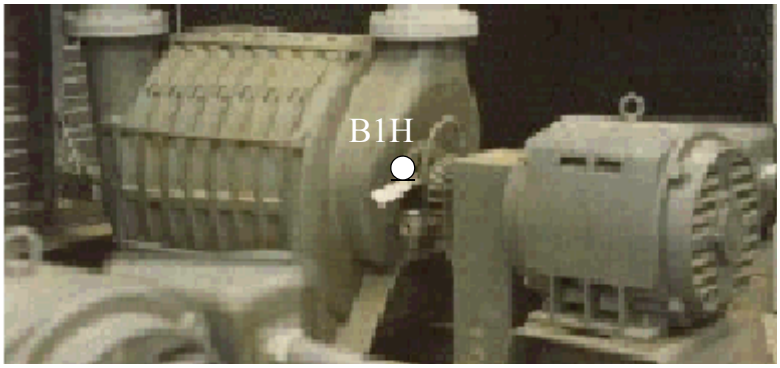


Figure 2

The spectrum in Figure 3 is the single spectrum for point B1H. Note the strong, single peak at 1xTS or one order. The high amplitude warrants corrective attention. The unbalance could lead to additional damage such as looseness, bearing failure, etc. Waveform analysis can help confirm the problem.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

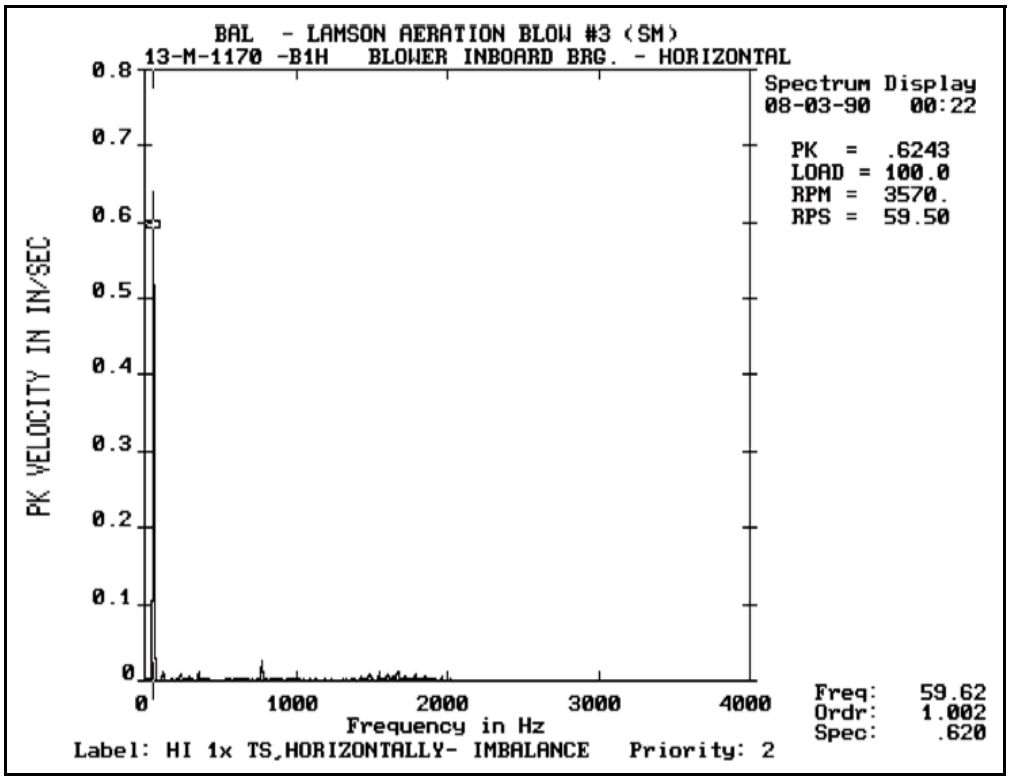


Figure 3

The waveforms in Figure 4 and 5 are from the same measurement point. Although not sinusoidal, Figure 4 has a discernible pattern. It was collected using digital integration, which allows data storage in the raw units of the transducer. Remember that acceleration accentuates, or amplifies, high frequencies. This characteristic aids in bearing defect detection, but it is not as useful for unbalance.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The waveforms in Figure 4 and 5 are from the same data point. Figure 3 waveform was stored using digital integration. Therefore the amplitude units are acceleration. The waveform is somewhat sinusoidal in nature, but also has some evidence of impacting in the sharp spikes. This helps in diagnosing potential rolling element bearing defects, and is useful in confirming a balance issue.

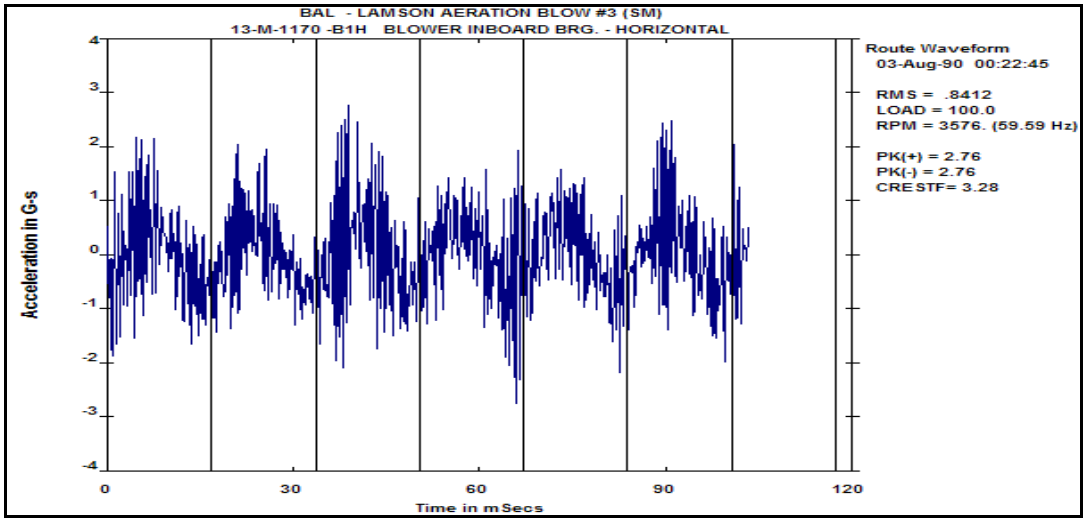


Figure 4

The waveform in Figure 5 is from the same point, but the amplitude units are velocity. The sinusoidal pattern is more pronounced. However, the serrated edges indicate the presence of some high frequency energy. This effect is useful for detecting an imbalance problem, but not for rolling element bearing defects.

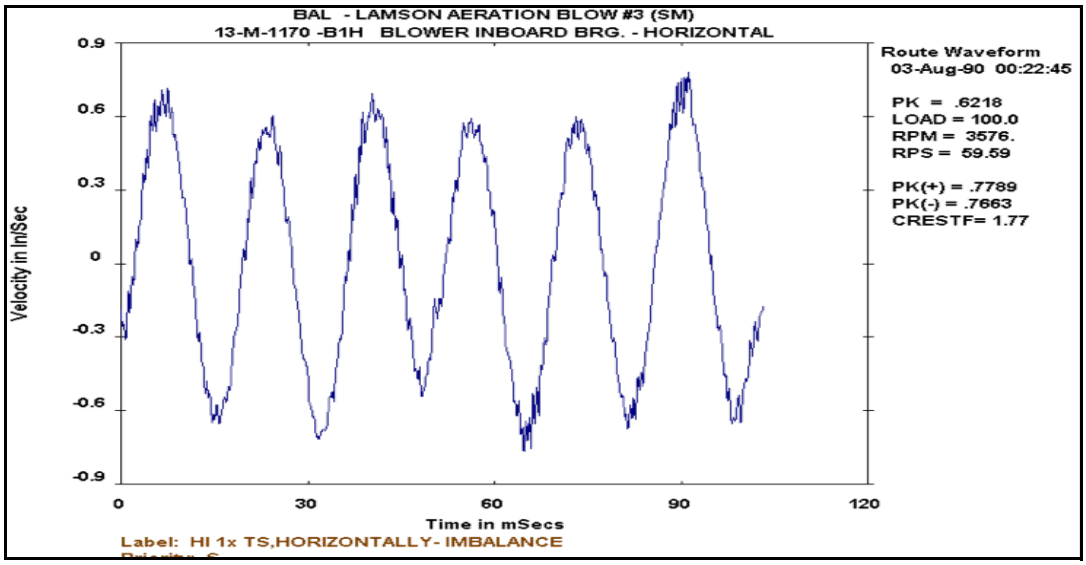


Figure 5

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Case History #2 - Turbine Driving ID Fan

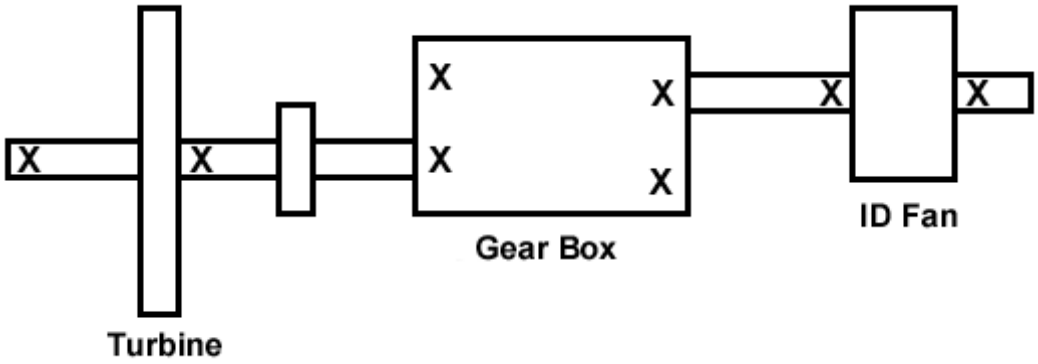


Figure 6

Equipment Data

- Bucket Type
- Approximately 900 HP
- Cast (not welded) design
- Speed reduced more than 1500 RPM due to excessive vibration at normal operating speed
- Prior failure due to seized coupling
- Missing bucket is suspected

Additional Notes:

- Turning speed harmonics are absent in the data from TIH measurement point
- Unbalance verified by other technicians
- Peak List included in case history Table 1

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

LIST OF SPECTRAL PEAKS
****

Machine: (BAL) TURBINE (DRIVING ID FAN)
Meas. Point: TURBINE -TIH --> TURBINE INBOARD HORIZONTAL
Date/Time: 11-12-87 14:20:14 Amplitude Units: IN/SEC PK

	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
	1	5.71	.0809	.13	13	438.67	.0058	10.14
	2	17.00	.0180	.39	14	452.34	.0052	10.46
	3	21.72	.0187	.50	15	457.66	.0045	10.58
	4	29.81	.0158	.69	16	476.28	.0054	11.01
)>	5	43.20	.3017	1.00	17	490.57	.0043	11.34
	6	52.52	.0089	1.21	18	495.69	.0048	11.46
	7	57.63	.0105	1.33	19	519.58	.0045	12.01
	8	69.18	.0079	1.60	20	527.85	.0046	12.20
	9	86.41	.0095	2.00	21	534.08	.0046	12.35
	10	91.83	.0047	2.12	22	538.70	.0057	12.46
	11	113.20	.0048	2.62	23	549.05	.0046	12.69
	12	129.44	.0048	2.99	24	560.64	.0053	12.96

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.3270	.1184 / 13%	.3010 / 85%	.0480 / 2%

Table 1

The data in Figure 7 shows the 1xTS peak as the highest amplitude peak on both the inboard (TIH) and outboard (TOH) locations. The corresponding vertical points are also exhibiting substantial TS amplitudes.

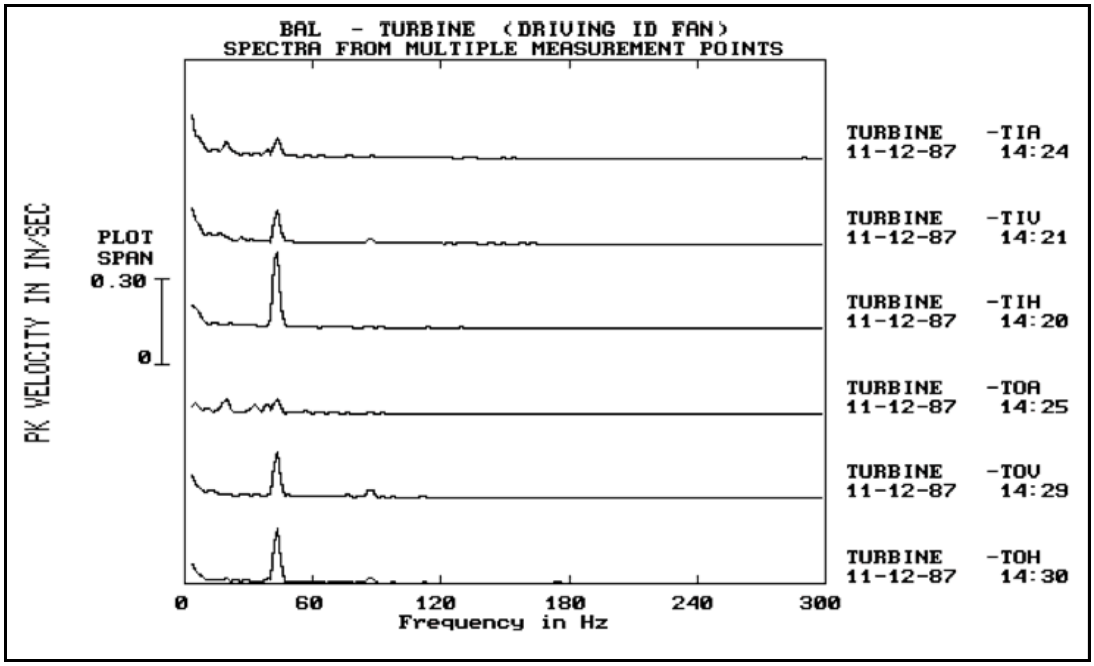


Figure 7

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

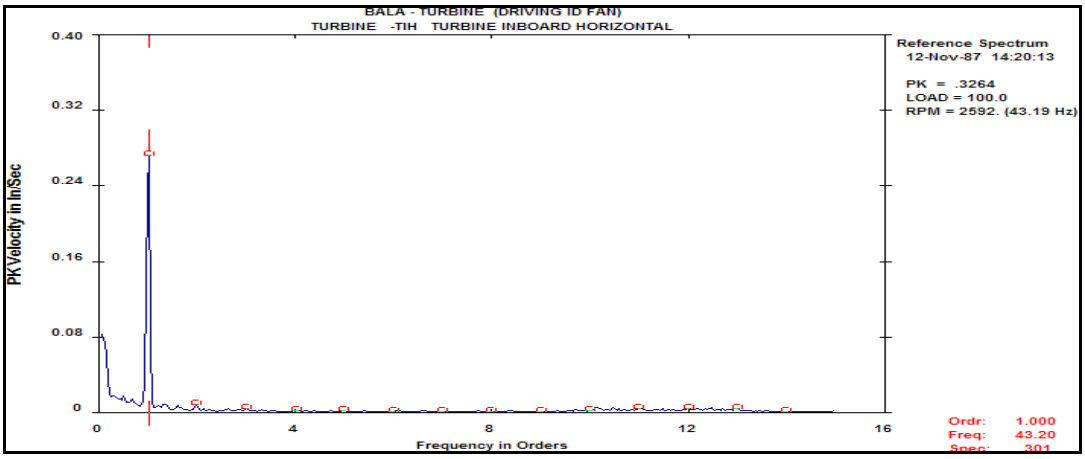


Figure 8

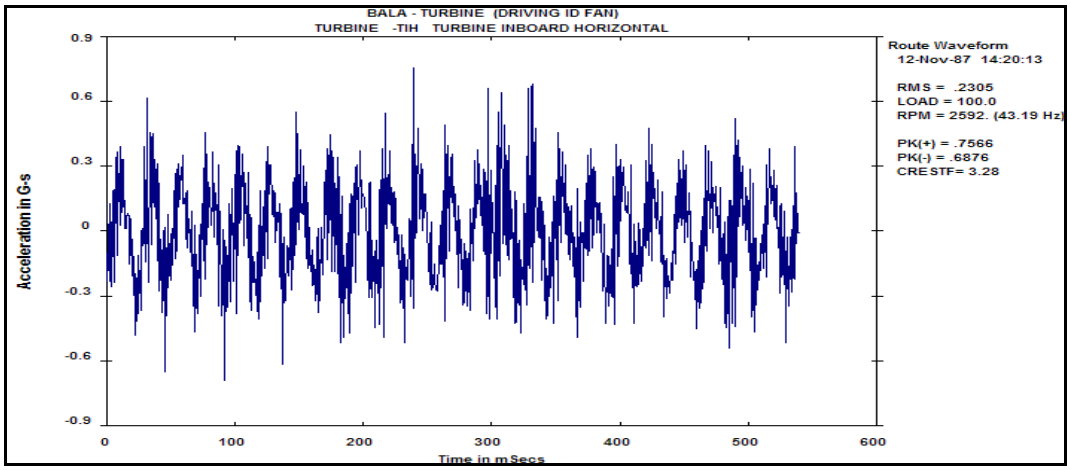


Figure 9

Data Analysis:

1) Vertically mounted machine (fasteners are vertical)
2) High amplitude turning speed peaks in the radial direction (Figure 8)
3) No turning speed harmonics (Figure 8)
4) Low amplitude TS peaks in the axial direction (Figure 8)
5) Periodic, sinusoidal waveform (Figure 9)
7) Period between peaks corresponding to turning speed (Figure 9)
8) Adjacent gearbox accounts for the high frequency data (Figure 6)

Diagnosis: Unbalance

Corrective Action: Balance Turbine Rotor

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Case History #3 - Coal Pulverizer

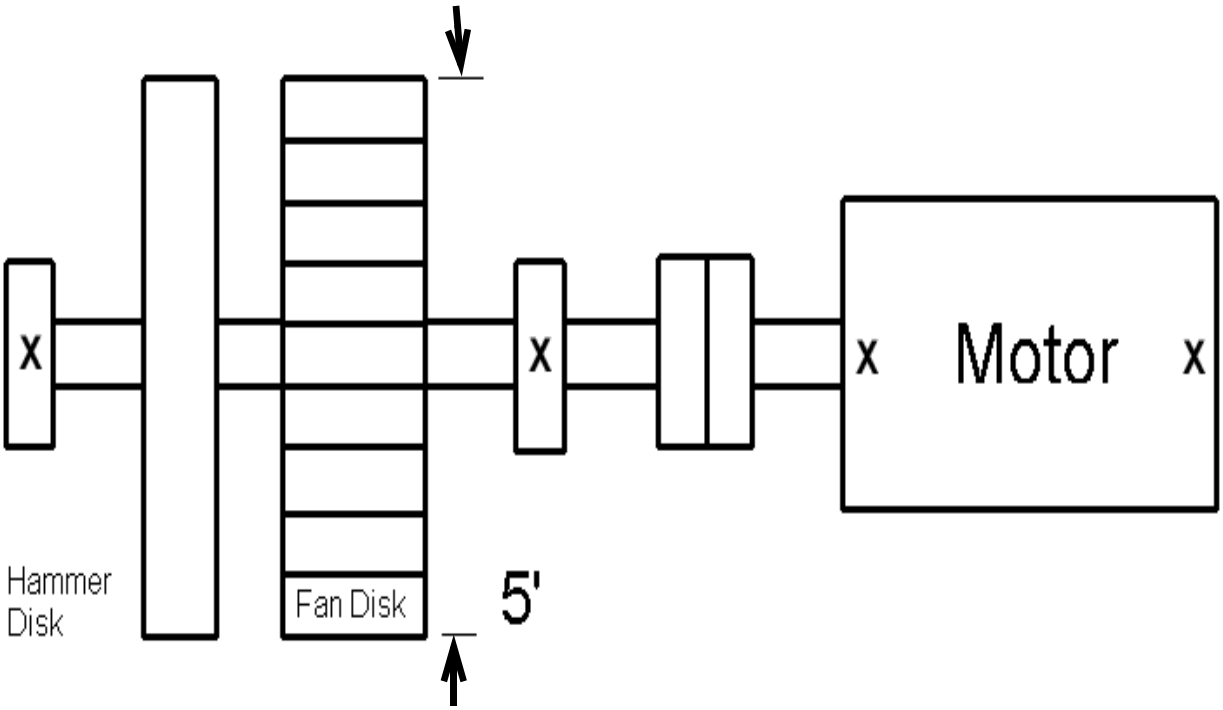


Figure 10

Equipment Data:

Pulverizer:	**Motor:**
Center Hung, double rotor unit	150 Horsepower
Outside rotor has hammers attached	6 Pole, Induction
Inside rotor has paddle-type fan blades attached	Turning speed: @ 19 Hz
Similar to a forced draft fan	

Additional Notes:

One of 14 pulverizers
As fan blades wear out, new ones are added
Two were undergoing re-builds
Standard procedure requires balancing prior to commissioning

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The multiple point plot in Figure 11 shows all three measurement positions on each of the two pulverizer bearings.

The inboard bearing positions are FIV, FIH, and FIA. The outboard bearing positions are FOV, FOH, and FOA. Note the relatively low axial vibration levels seen in FIA and FOA.

The vertical readings - FIV and FOV - are also low, probably because of the vertical stiffness of the bearings.

The horizontal readings - FIH and FOH - are both high and of similar magnitude - over 0.8 and 0.6 IPS. Very little harmonic activity appears in any of the six measurements.

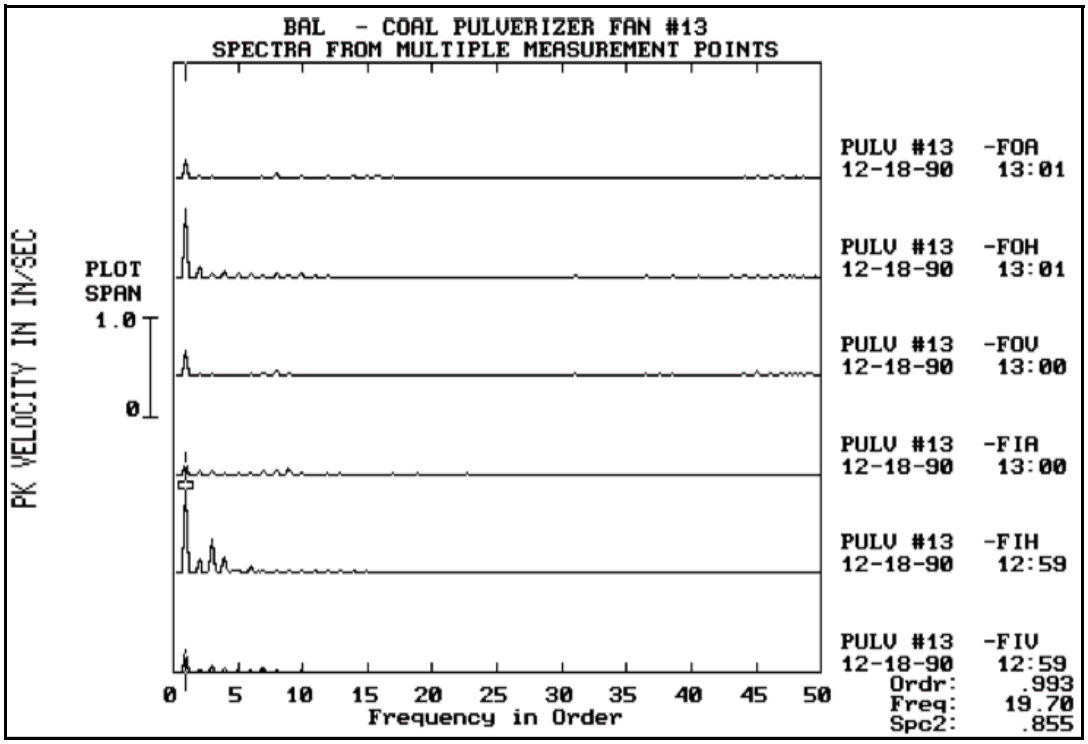


Figure 11

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

A single-spectrum view of FOH in Figure 12 reveals a major 1xTS peak. The harmonic peaks of turning speed are relatively insignificant. The harmonics are probably caused by the 1XTS vibration shaking the entire structure.

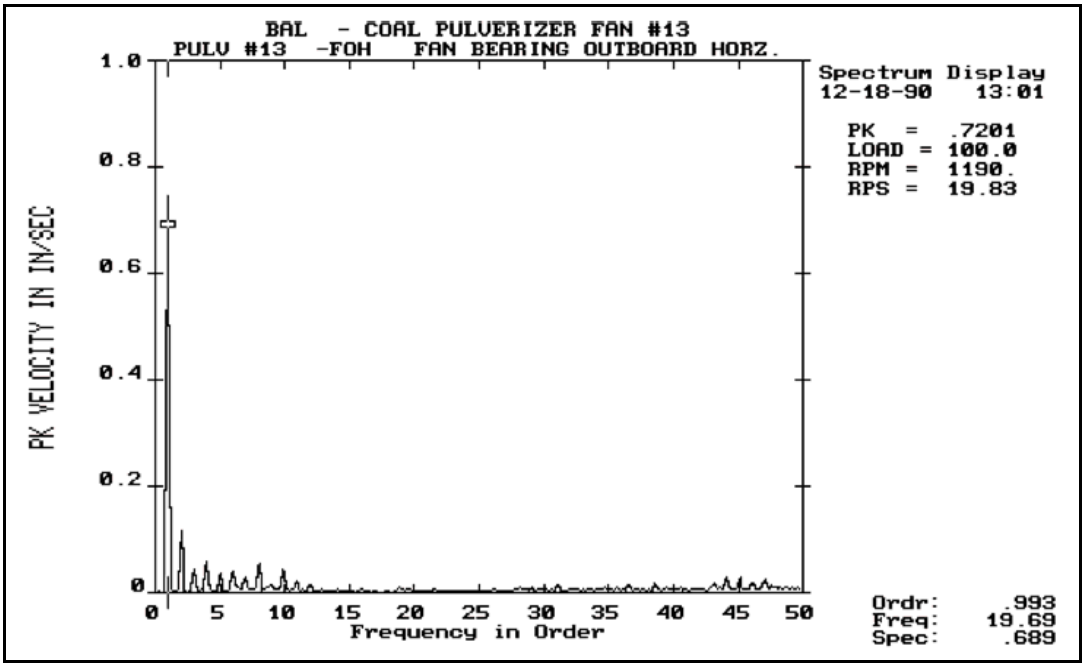


Figure 12

LIST OF SPECTRAL PEAKS
****

Machine: (OFLT) COAL PULVERIZER
Meas. Point: PULV 13 -BOH --> BEARING OUTBOARD HORZ.
Date/Time: 18-Dec-90 13:01:19 RPM= 1190. Units=In/Sec PK
Data Label: OUT BALANCE SHAFT & BLADING

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	19.69	.6867	.993	13	615.77	.0170	31.047
2	39.38	.1184	1.986	14	725.20	.0164	36.565
3	59.06	.0451	2.978	15	764.52	.0212	38.547
4	78.72	.0611	3.969	16	854.35	.0183	43.076
5	98.43	.0366	4.963	17	874.55	.0292	44.095
6	118.17	.0418	5.958	18	894.07	.0314	45.079
7	137.77	.0318	6.946	19	913.81	.0218	46.074
8	157.66	.0609	7.949	20	933.62	.0268	47.073
9	177.19	.0159	8.934	21	943.09	.0154	47.551
10	196.75	.0456	9.920	22	950.47	.0132	47.923
11	216.72	.0245	10.927	23	965.35	.0134	48.673
12	236.60	.0163	11.929	24	982.72	.0136	49.549

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.7124	0.0 / 0%	.6976 / 96%	.1446 / 4%

(ELECTRICAL= 0.00E+00/ 0% FLOOR= .0762 SPEC CREST= 3.15)

Table 2

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The time waveform that generated the spectrum in Figure 12 appears in Figure 14 below. The 1xTS peak that dominates the spectrum indicates not only that the waveform should appear sinusoidal, but also that the time spacing should equal the frequency of the 1xTS peak.

The vertical lines on the waveform represent the time required for the shaft to make one revolution. One major peak clearly marks each revolution of the shaft.

The waveform looks very periodic but not complex in nature.

When analyzing a waveform, be sure to note the amplitude units. This waveform is in velocity. A velocity waveform is especially useful in confirming low frequency events such as imbalance. The ragged edge in this data indicates the possibility of some higher frequency events occurring.

If Digital integration had been implemented, the waveform would have been in acceleration.

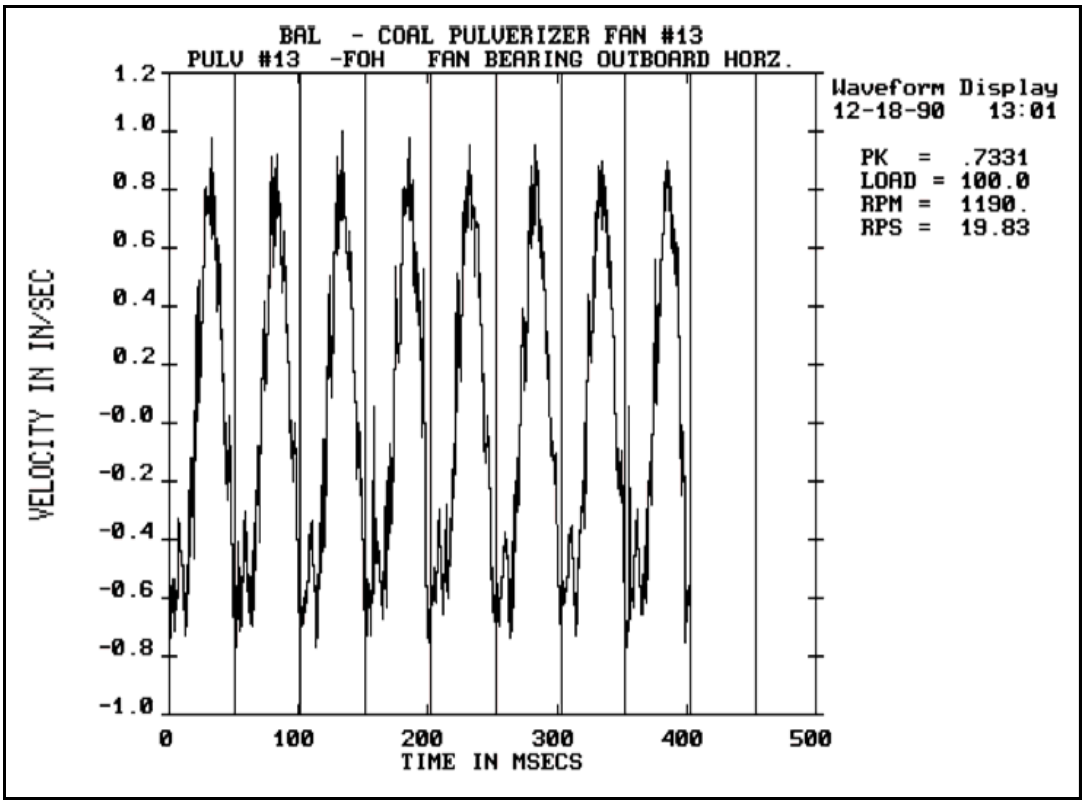


Figure 14

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Due to the inaccessibility of the outboard (hammer) disk, this balance job called for a single plane application. In this case, accelerometers were mounted in the two horizontal positions on the fan bearings. The first balance shot of 41 ounces brought the vibration down to the levels illustrated in Figure 15.

The six measurement points shown in Figure 15 show the data collected after the unit was balanced. Note the amplitude scale decreased from a full scale range of 1.0 IPS (Figure 11) to 0.2 IPS (Figure 15). No single peak exceeds an amplitude of 0.2 IPS.

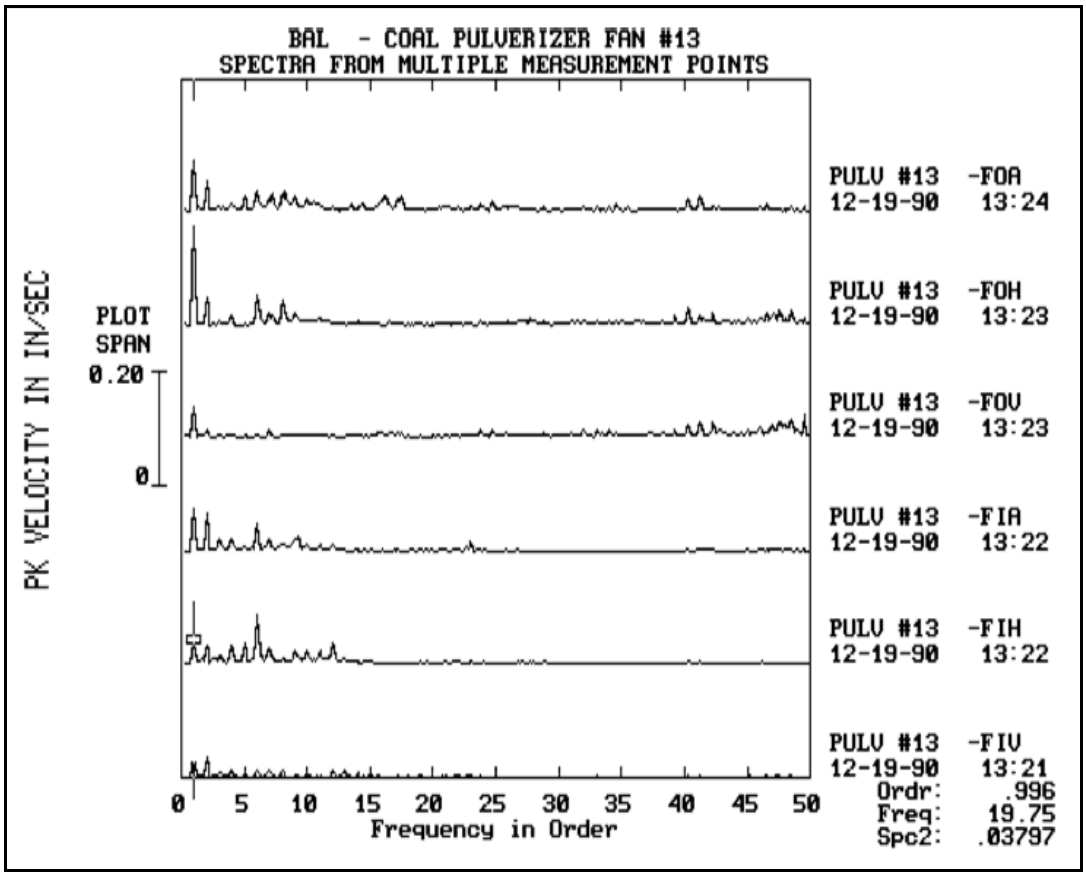


Figure 15

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

A single-spectrum view of FOH appears in Figure 16. The peak at 1xTS, while still dominant, has decreased from over 0.8 IPS to less than 0.2 IPS. Note the hump of energy now visible between 5xTS and 10xTS.

The waveform has become relatively more complex, thereby causing the hump of energy. This kind of hump also adds significant energy to the overall spectrum vibration level.

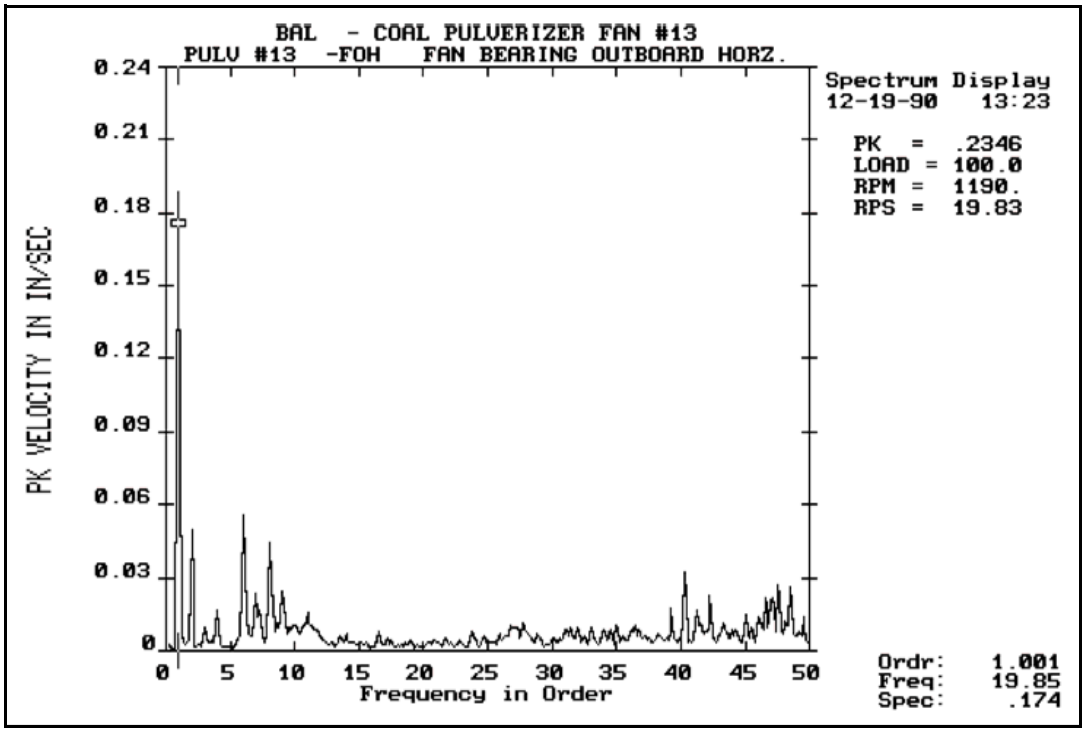


Figure 16

Data Analysis:

1) Vertically Mounted
2) Dominant Synchronous Energy
3) High Amplitude Turning Speed Peak (Figure 12)
4) Low Amplitude Turning Speed harmonics (Figure 15)
5) Periodic, sinusoidal waveform (Figure 14)
6) Less than 2g swing (Figure 15)
7) Relatively Low Axial energy (Figure 16)

Diagnosis: Unbalance
Corrective Action: Balance Fan Disk

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The time waveform in Figure 17 after the balance shot shows much lower amplitude ±0.3 IPS instead of ±0.8 IPS. The shape remains periodic, although the waveform has become more complex. The complex, random energy causes the energy hump seen in the spectrum between 5xTS and 10xTS.

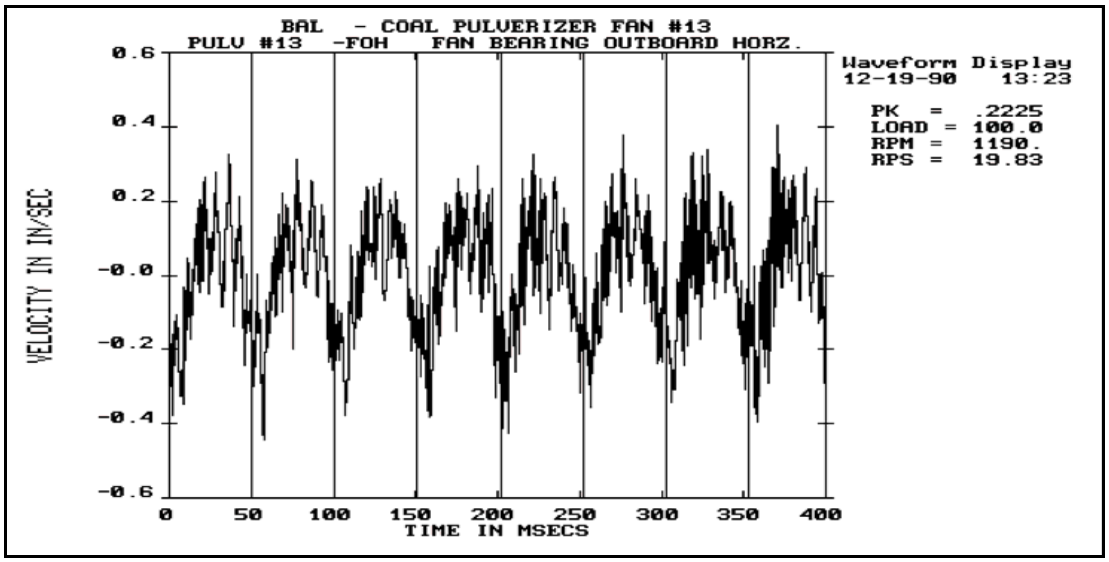


Figure 17

Spectra from before and after the balance job appear in Figure 18 to show the significant difference in the vibration. Vibration Levels were reduced in all three directions. This machine pulverizes large blocks of coal. Vibrations of this amplitude are probably acceptable.

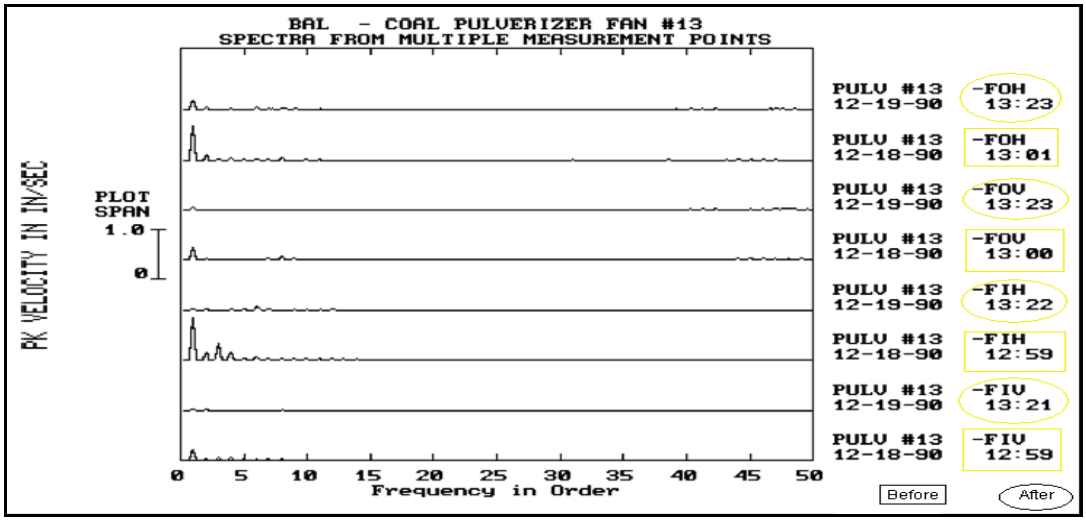


Figure 18

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Case History #4 - Reactor Fan #6

200 HP motor
Direct Drive Center Hung
Squirrel Cage Fan

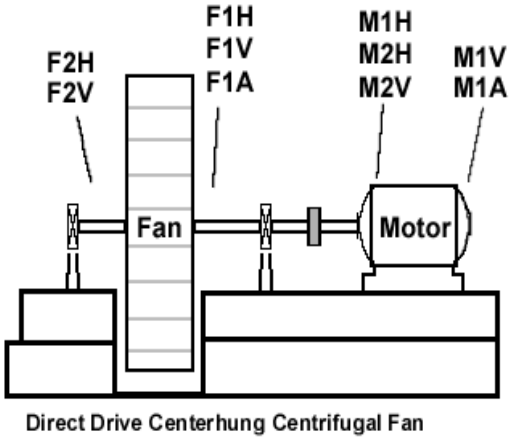


Figure 19

Figure 20 is a **Multiple Points Plot** of the Fan measurements points. Notice that most of the vibration is at 1xrpm in the horizontal direction

.

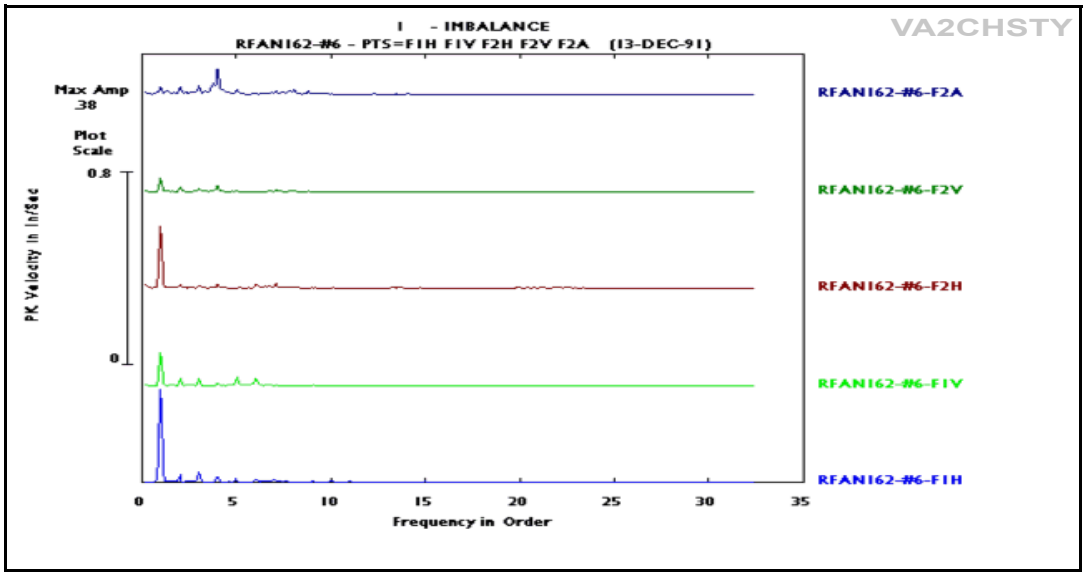


Figure 20

Unbalance is manifest more clearly on a vertically mounted machine.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The peak list in Table 3 shows that the majority of energy is synchronous in nature.

LIST OF SPECTRAL PEAKS
****

Machine: (BAL) REACTOR FAN - #4 (162) F1H
Meas. Point: RFAN162-#4-F1H --> FAN BEARING #1 - HORIZONTAL
Date/Time: 17-Jan-92 10:23:56 RPM= 1175. Units=In/Sec PK
Data Label: imbalance

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	4.47	.0067	.228	13	130.94	.0067	6.686
2	7.25	.0029	.370	14	137.68	.0051	7.030
3	19.70	.4754	1.006	15	150.21	.0025	7.670
4	29.16	.0055	1.489	16	157.76	.0070	8.056
5	32.52	.0049	1.661	17	177.41	.0068	9.059
6	39.45	.0193	2.014	18	217.05	.0032	11.083
7	45.55	.0040	2.326	19	236.49	.0075	12.076
8	53.15	.0061	2.714	20	239.41	.0036	12.225
9	59.14	.0440	3.020	21	247.76	.0027	12.652
10	78.89	.0199	4.028	22	251.76	.0025	12.856
11	98.48	.0163	5.029	23	296.45	.0026	15.138
12	118.30	.0129	6.041	24	315.49	.0053	16.110

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.4792	.0073 / 0%	.4768 / 99%	.0470 / 1%
(ELECTRICAL=	0.00E+00/ 0%	FLOOR= .0211	SPEC CREST= 1.32)

Table 3

The spectrum from the Fan Inboard Horizontal position in Figure 21 reveals elevated **Overall** amplitude at almost 0.5 in/sec. Turning speed is located with an amplitude of 0.475 in/sec. Almost all of the energy is being generated by the turning speed of the machine.

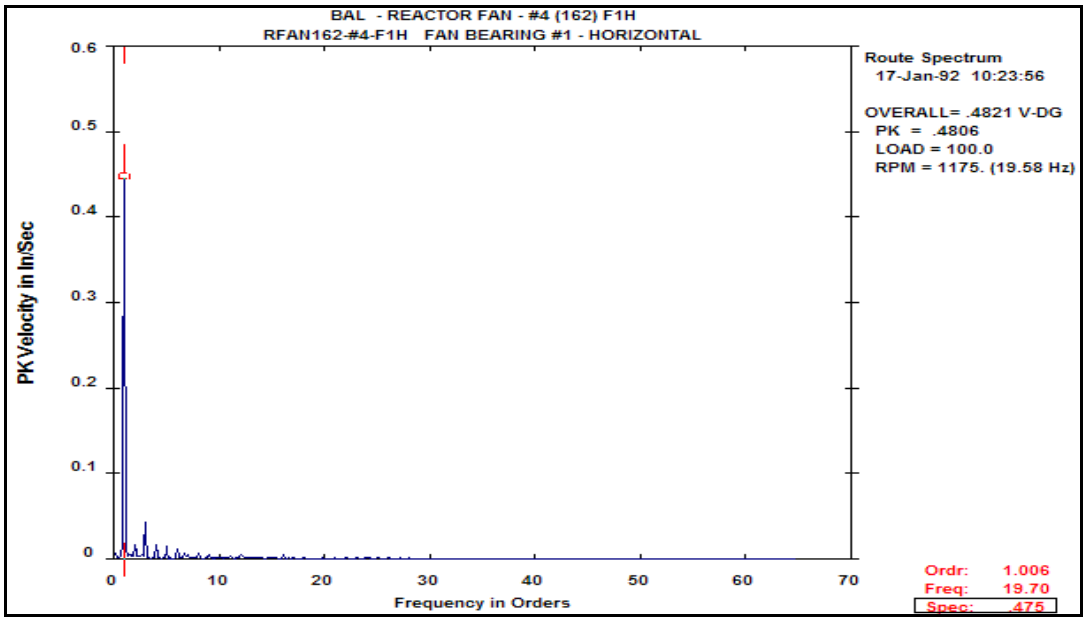


Figure 21

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The waveform from F1H in Figure 22 has a sinusoidal, one event per revolution pattern.

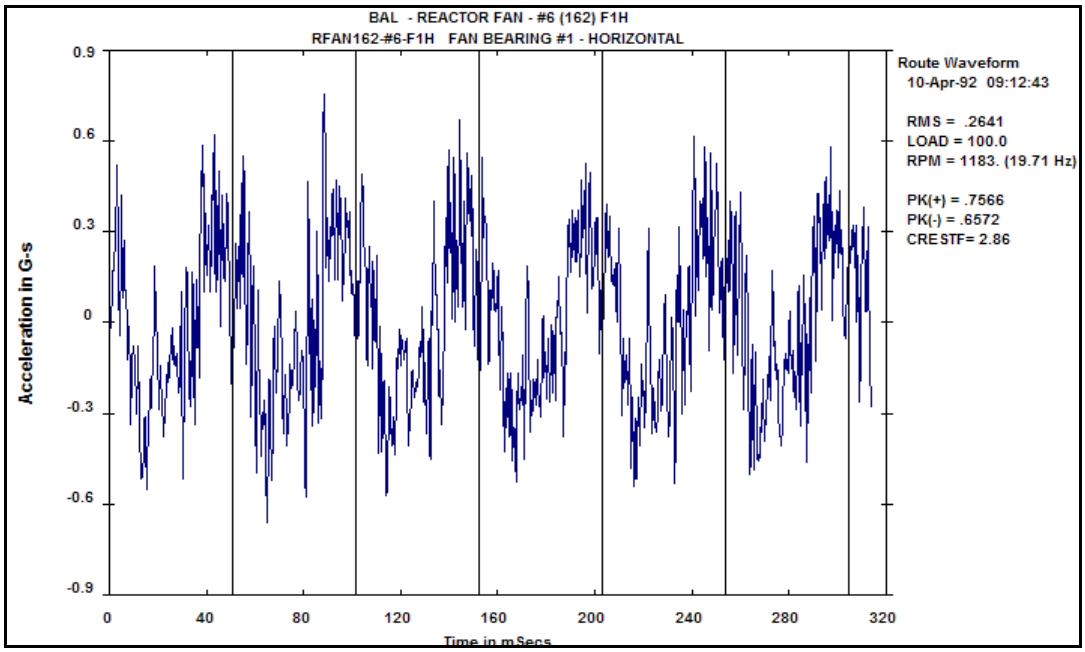


Figure 22

The data from the axial direction in Figure 23 indicates low amplitude turning speed energy.

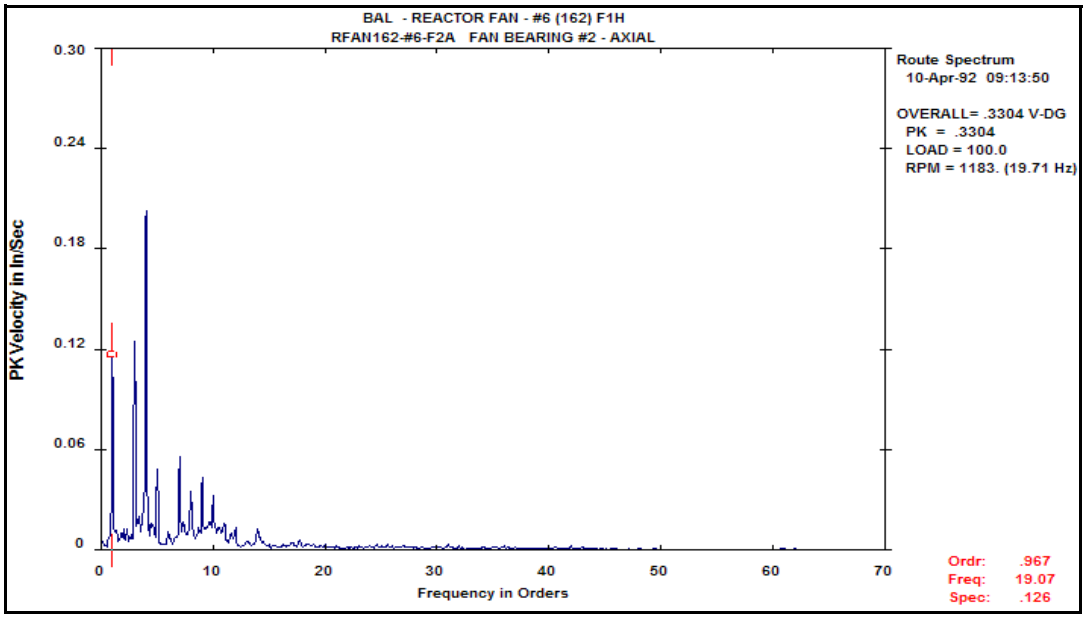


Figure 23

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The single spectrum of the Fan Inboard Horizontal point is in Figure 24. Notice the high **Overall** amplitude of vibration in the Horizontal direction. Overall vibration is almost 0.9 in/sec.

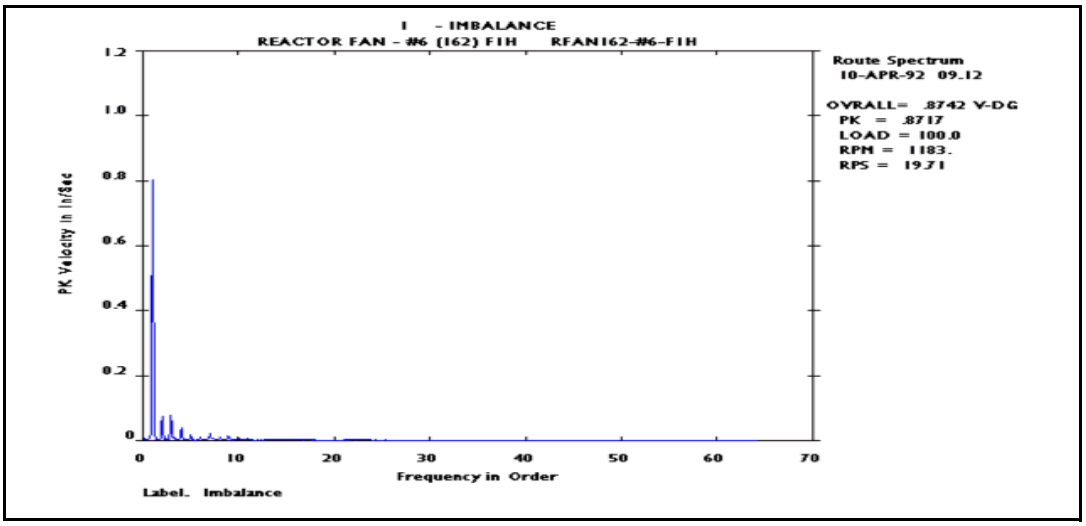


Figure 24

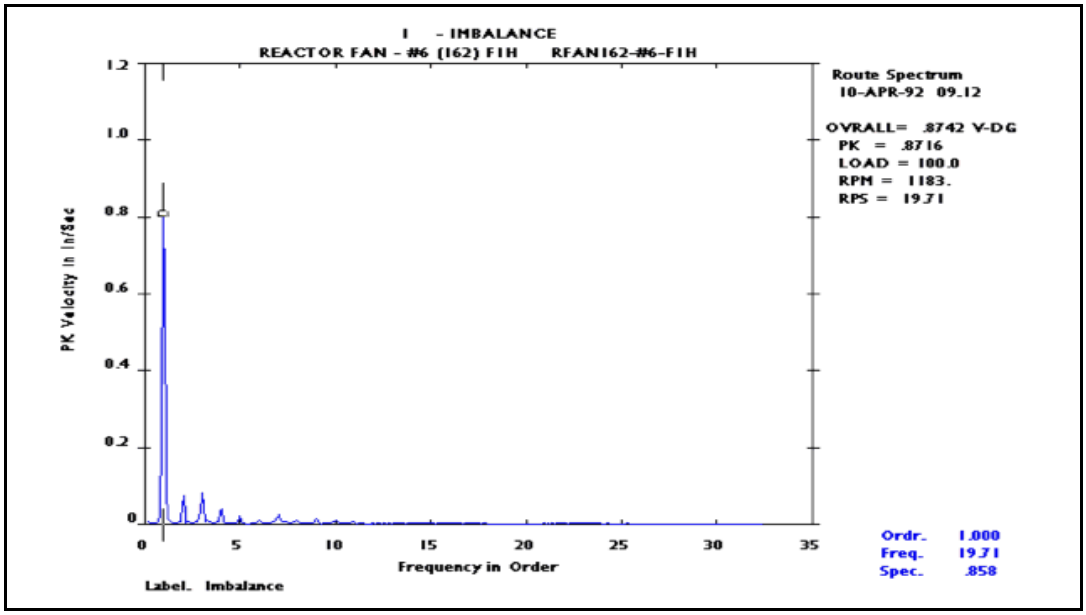


Figure 25

In Figure 25, the cursor is marking 1xrpm. Notice in the lower right hand corner, the spectral amplitude for 1xrpm is 0.858 in/sec. Almost all of the vibration on this machine is at 1xrpm in the Horizontal direction, indicating unbalance.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The Time Waveform from the FIH measurement point is a sinusoidal 1x per revolution type pattern. This is another indication of unbalance.

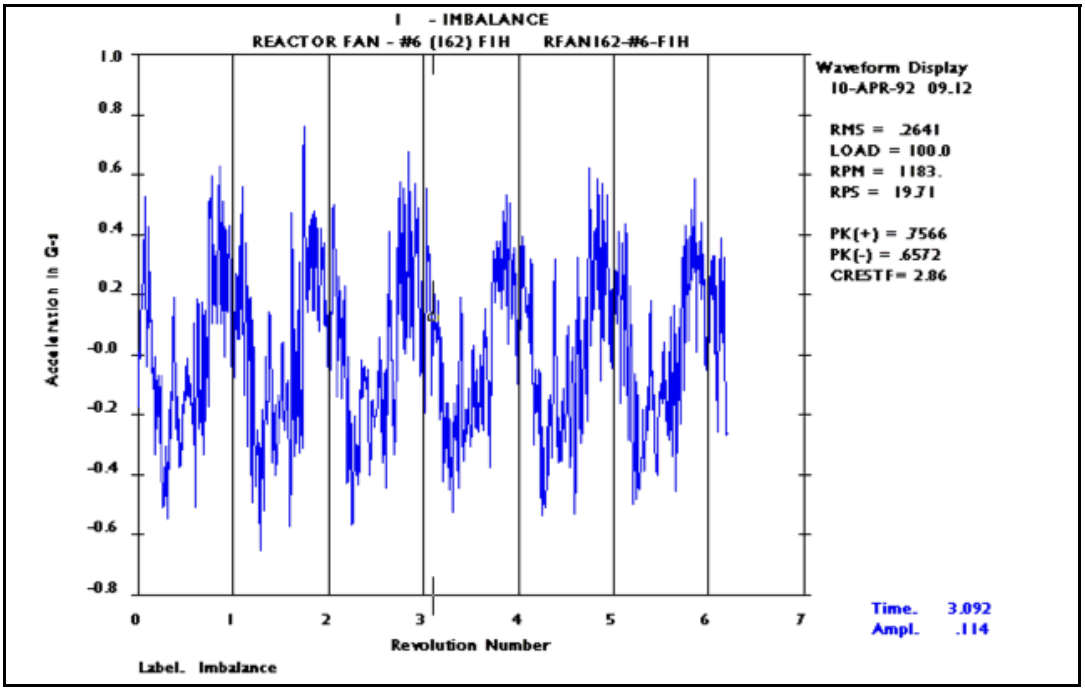


Figure 26

Data Analysis:

1) Vertically mounted machine (fasteners are vertical)
2) Predominantly synchronous energy (Table 3)
3) High amplitude turning speed peaks in the radial direction (Figure 21)
4) Low amplitude turning speed harmonics (Figure 21)
5) Low amplitude TS peaks in the axial direction (Figure 23)
6) Periodic, sinusoidal waveform (Figure 22)
7) Less than 2 g swing - little impacting (Figure 22)
8) Period between peaks corresponding to turning speed (Figure 22)

Diagnosis: Unbalance

Corrective Action: Balance Fan

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The data displayed in Figure 27 is of the Fan Inboard Horizontal point after the fan was balanced. Notice the **Overall** amplitude and the amplitude at 1xrpm.

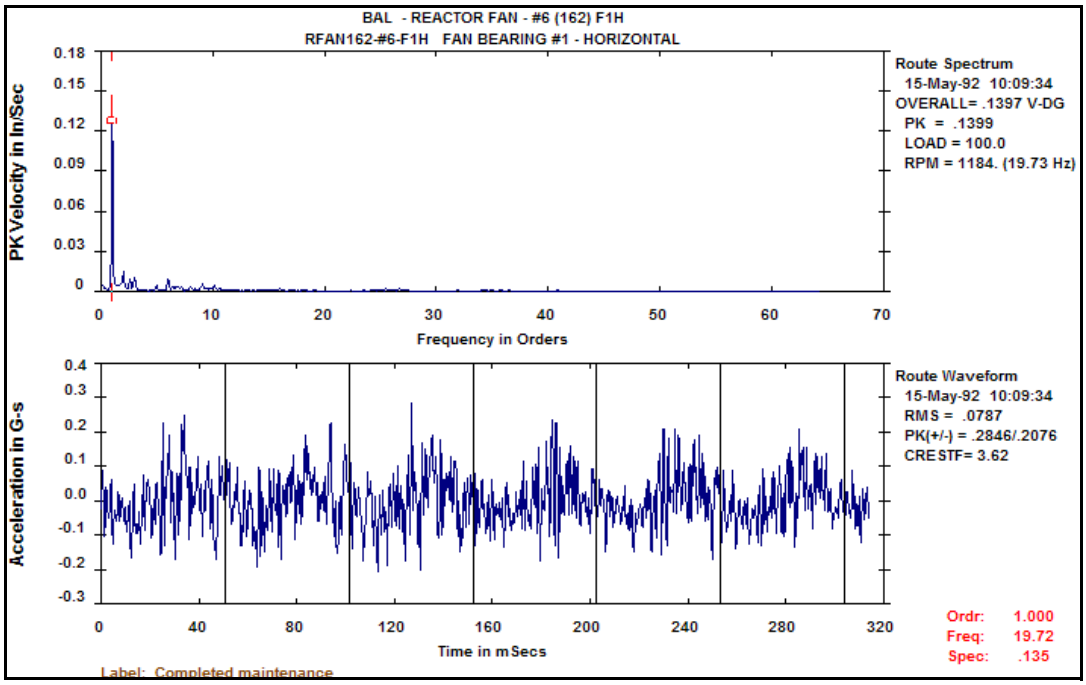


Figure 27

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Case History #5 - Combustion Air Fan

Unbalance with Looseness

The Spectrum in Figure 28 is from an unbalanced machine in which the unbalanced condition is causing looseness. Turning speed is marked with the primary cursor and the harmonics of turning speed are marked with the harmonic cursors.

This is a good example of what happens when an unbalanced condition is not corrected. Unbalance will eventually lead to other problems such as looseness, bearing defects, leaking seals, and misalignment. The turning speed peaks and harmonics will probably diminish significantly when the machine is balanced.

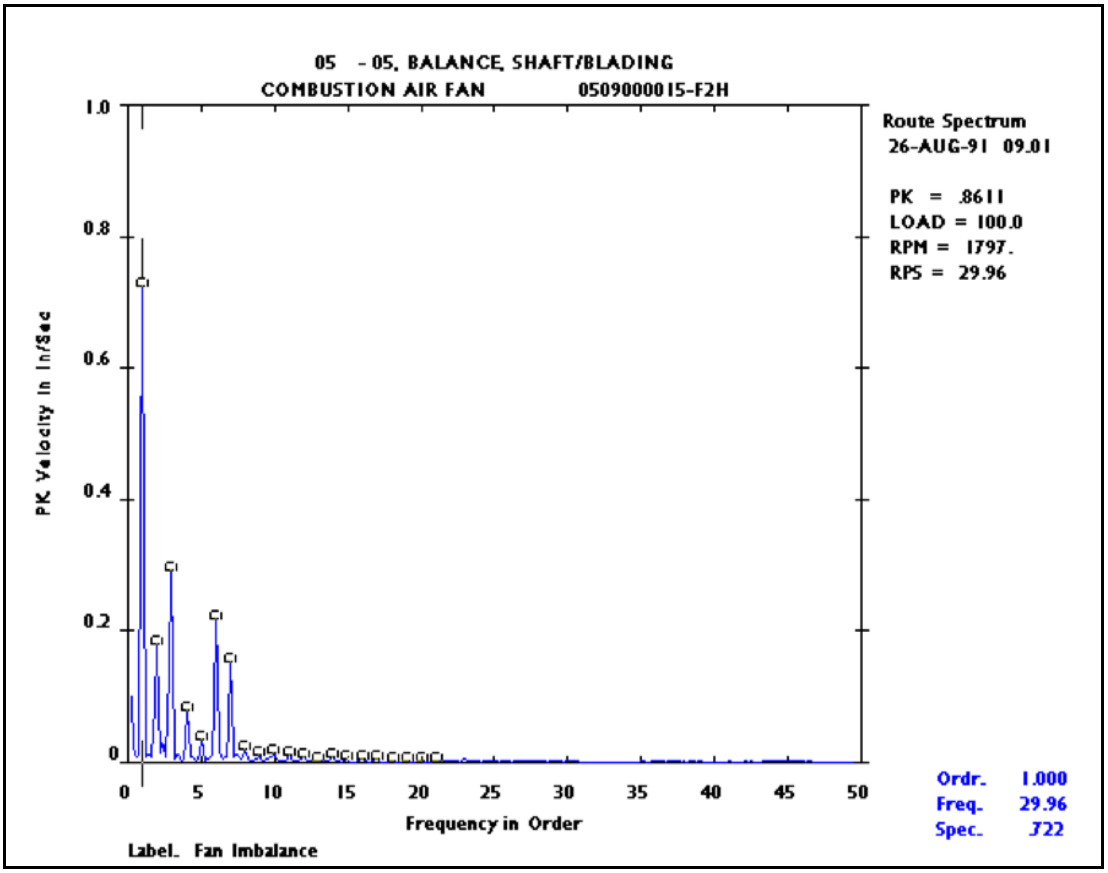


Figure 28

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

The Time Waveform also shows evidence of the Unbalance and Looseness. The harmonic cursors are marking the rotational frequency in the waveform. The other peaks represent the Looseness per revolution. (Figure 29)

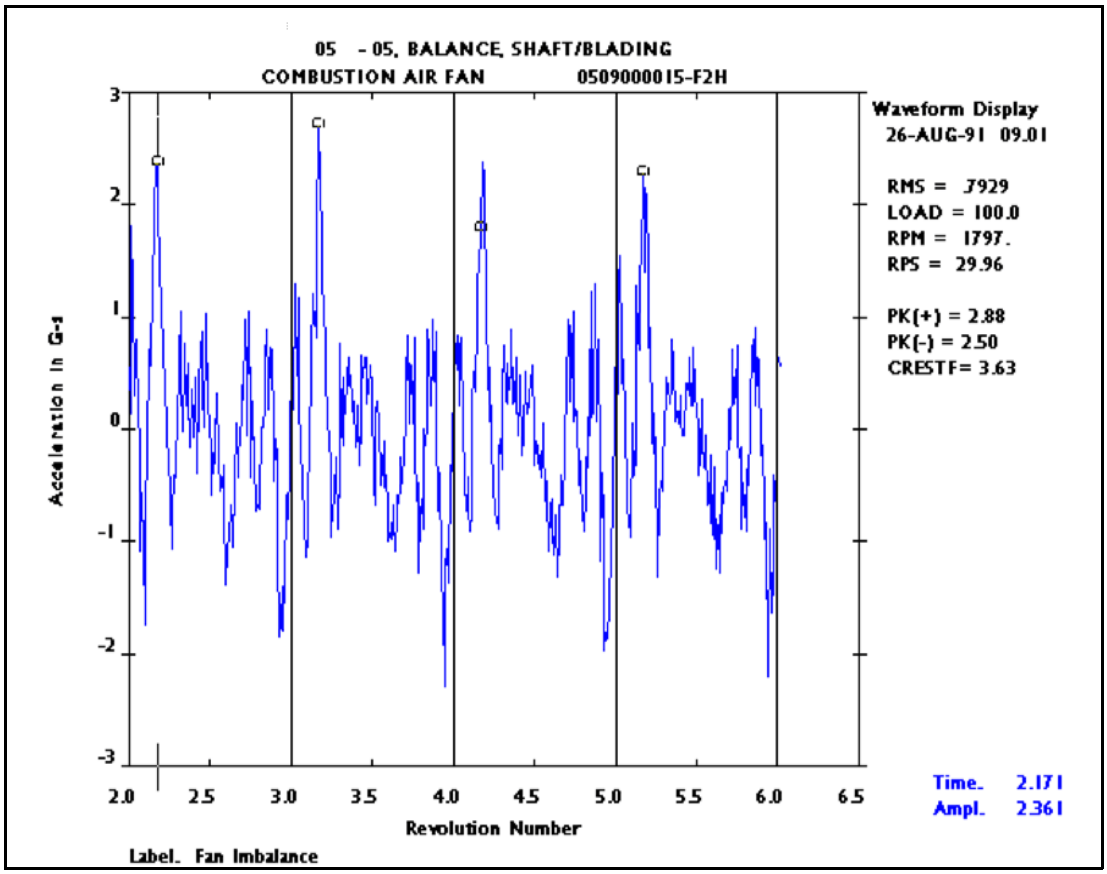


Figure 29

Note

The Harmonic cursors may not always match up perfectly in the Waveform. This can be due to slight variations in speed or sampling rate.

Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

 Copyright 2006. Emerson Process Management. All Rights Reserved 04/06

Misalignment

Section 3

Objectives

- Define Misalignment.
- Determine some causes of misalignment.
- Identify the characteristics of misalignment.
- Establish some corrective actions.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Misalignment

Misalignment is the condition where two connected shafts are either not parallel or do not share a common axis.

Belt drive equipment shafts should be parallel, direct coupled equipment should share a common axis when normal operating conditions, such as speed, load and temperature, are reached.

The three basic types of misalignment are:

- Offset
- Angular
- Bearing

Some common causes of shaft misalignment are:

- Improper base preparation
- Machine soft foot
- Pipe strain
- Improper training and tools
- Extreme thermal activity

Some characteristics of misalignment:

- High axial energy with angular misalignment
- Elevated radial energy for offset misalignment
- 180° Phase shift across the coupling

Waveform:

- Periodic, sinusoidal
- One or two events per revolution

Spectrum:

- Increased amplitudes of 1X and/or 2X peaks
- Possible elevated 3X with locked or damaged coupling
- Rule of thumb:
  If the 2X peak is 50% or greater in amplitude than 1X, misalignment is very possible.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Note

Pure shaft or coupling misalignment is a major source of excessive vibration. It can be remedied with proper training and resources. However, it can be very difficult to diagnose. Since misalignment may appear at 1x and 2x, or a combination of both, frequently the problem is treated as unbalance or bent shaft. Phase measurements are a vital part of analysis when misalignment is suspected. In many cases phase is the only conclusive evidence that will confirm misalignment. However, factors such as shaft diameter, speed of machine, equipment rigidity, critical speeds, type and diameter of coupling, operating temperatures, etc., can all have a detrimental effect on vibration amplitudes and phase data. Often the only way to confirm misalignment is to shut the machine down and perform an alignment check on it.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Misalignment- Types and Descriptions

Angular:

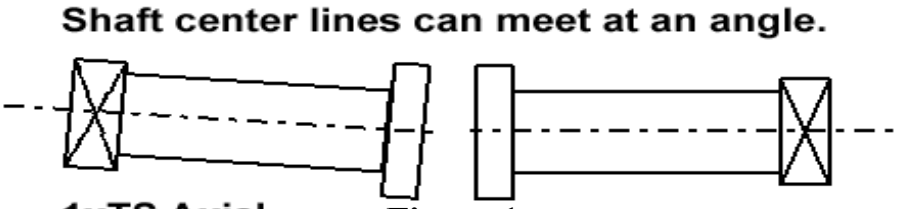


Figure 1

Separate Axes

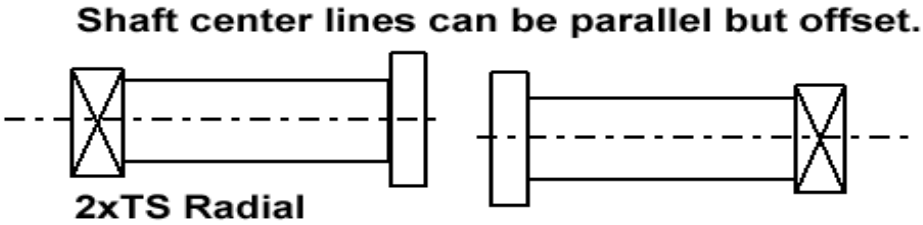


Figure 2

Combination of offset and angular:

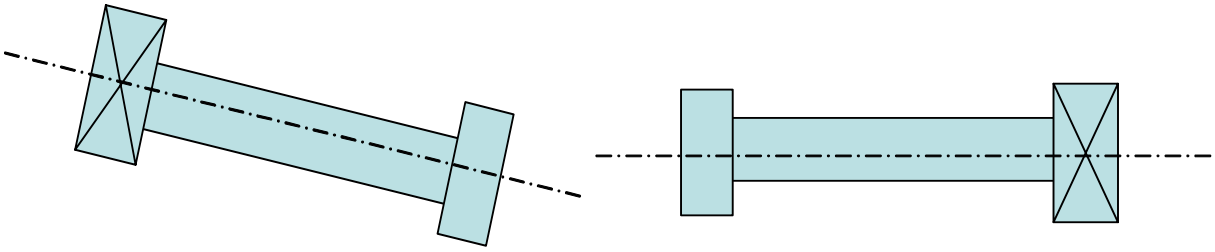


Figure 3

Possible elevated 1X and 2X radial and axial positions

Combination angular and offset:
1xTS - axial
2xTS - axial
1xTS - radial
2xTS - radial
1 xTS and 2xTS - radial

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Bearing misalignment in rotating machinery causes the shaft to bend as it passes through the end bells as shown in Figure 4. This condition causes high axial loads on the bearings and high axial vibration at 1xTS and 2xTS.

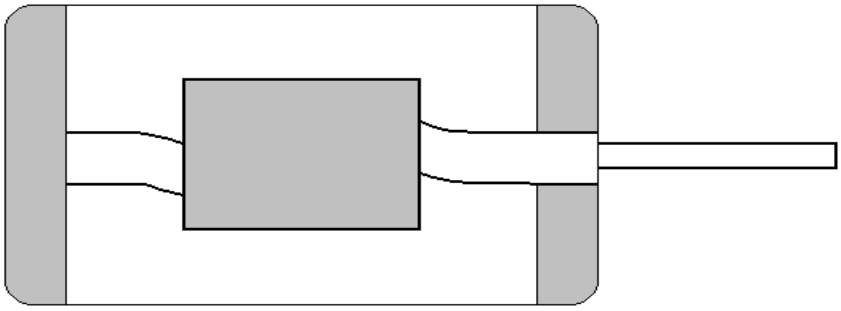


Figure 4

This condition may be detected using phase analysis on the face of the endbell. If bearing misalignment is present there should be approximately 90 degrees shift between measurement locations that are spaced 90 degrees apart. (Figure 5)

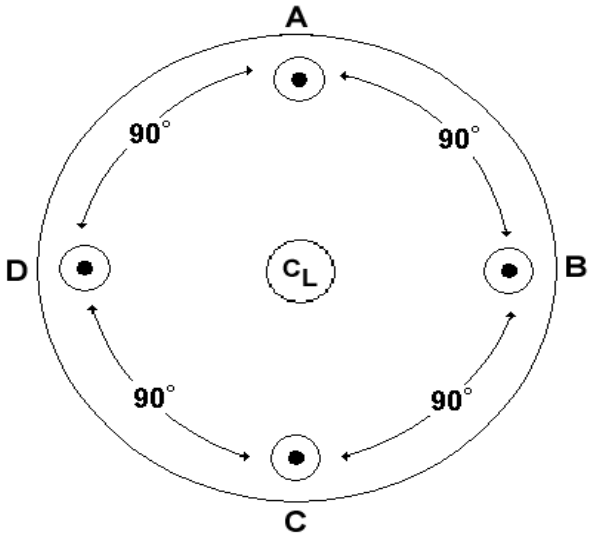


Figure 5

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #1 - Line shaft Turbine

Equipment Notes:

The shaft has been properly balanced. If the operating parameters are normal, it is not necessary to interrupt production for repairs. This turbine is shut down every two years for preventative maintenance. Critical components such as packing, seals, and couplings are inspected and replaced as necessary. Sleeve bearings support the turbine shaft, constituting a degree of looseness. Data was collected for re-certification purposes before commissioning.

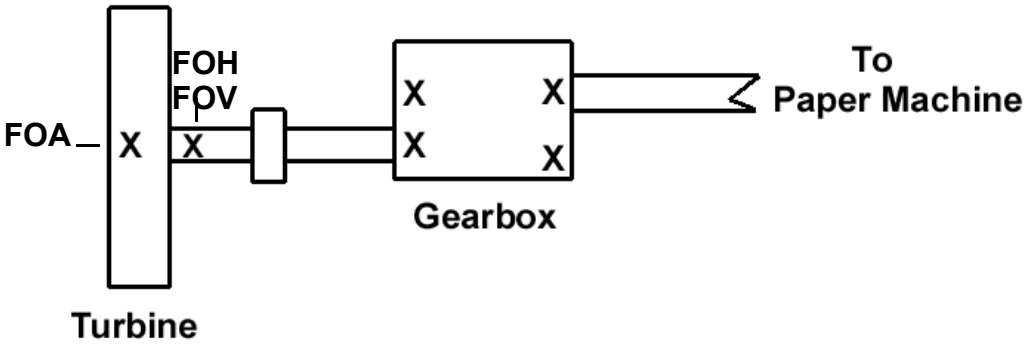


Figure 6

Other items to note include:

◆ The shaft has been well balanced.
◆ If the turbine operates normally now, it is not necessary to bring it down. This turbine is brought down at two year intervals. After two years this turbine could blow packing, damage seals, fail couplings, etc.
◆ The large 3X peak on point TOH suggests the possibility of looseness, or a locked coupling. TOH waveform helps confirm misalignment diagnosis.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The Multiple Point plot in Figure 7 indicates 1xTS harmonics on the TOV and TOH positions. It is normal to see some turning speed harmonics in sleeve bearing machines. However, there is a significant 2xTS peak in the data. An elevated 3xTS peak may be attributed to a worn or locked coupling. There is little “floor noise” in the spectra. The peaks are relatively crisp, clear and narrow. Broad based “skirts” in the spectrum could be a result of impacting in the waveform.

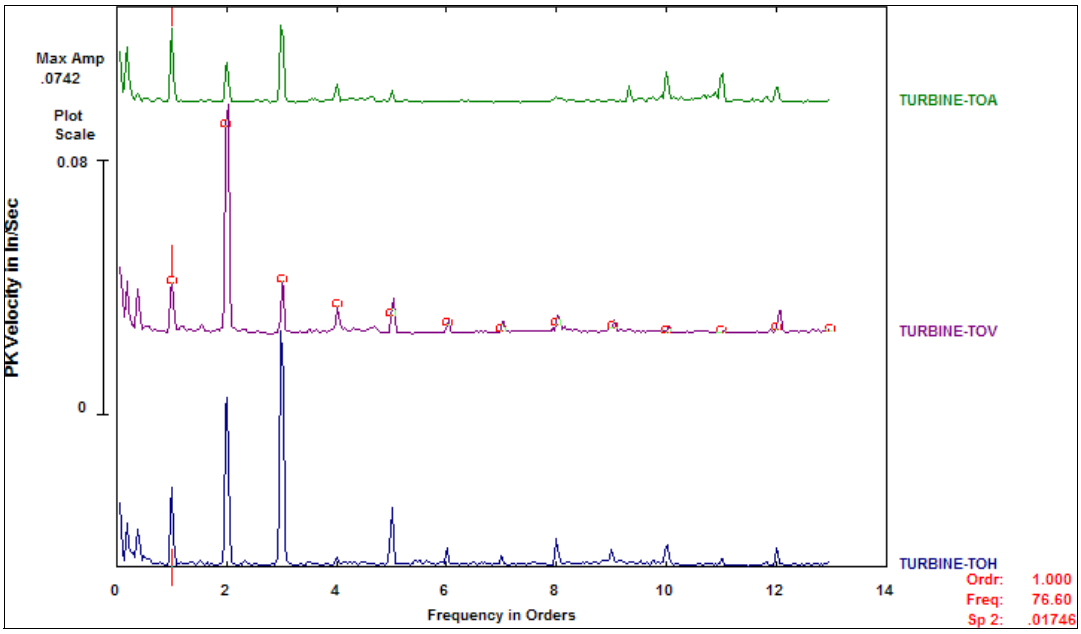


Figure 7

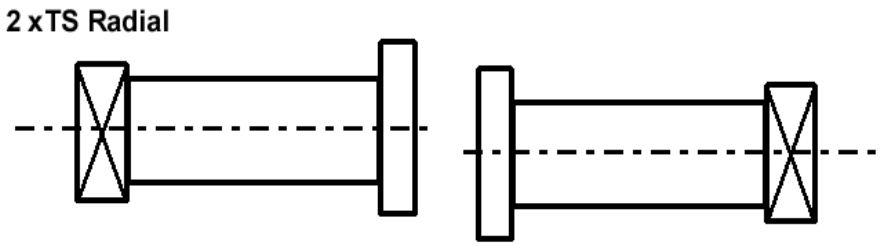


Figure 8

Reminder:

There are many variables involved when considering misalignment as the cause of excessive vibration. Speed, shaft diameter, coupling type, type of misalignment, etc., are some. Different combinations of these variables can affect amplitude and frequencies.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The single spectrum from the TOV position is showing a significant 2xTS peak, along with the 1xTS harmonics. Note the ratio between the 2xTS and the 1xTS peaks. The general rule is; if 2x is 50% (or more) of 1x, misalignment is present. In this case the ratio is much greater than that. 2x is almost 5 times the amplitude of 1x. (Figure 9)

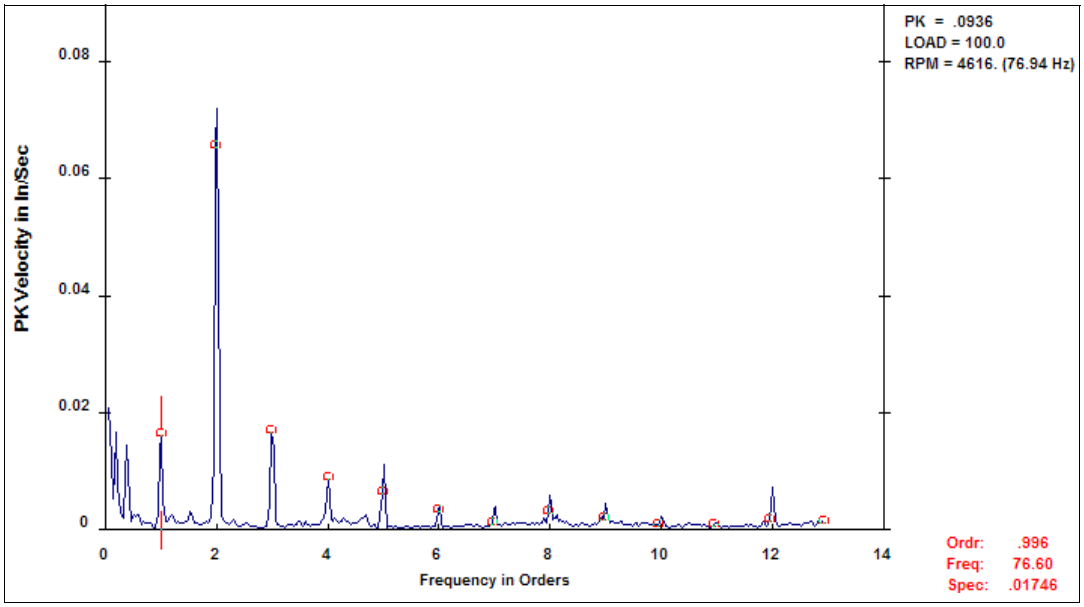


Figure 9

Normalizing frequency of the shaft is marked with vertical lines. The waveform is showing two distinct events per shaft revolution. The low 'g' level(s) indicate very little or no impacting.

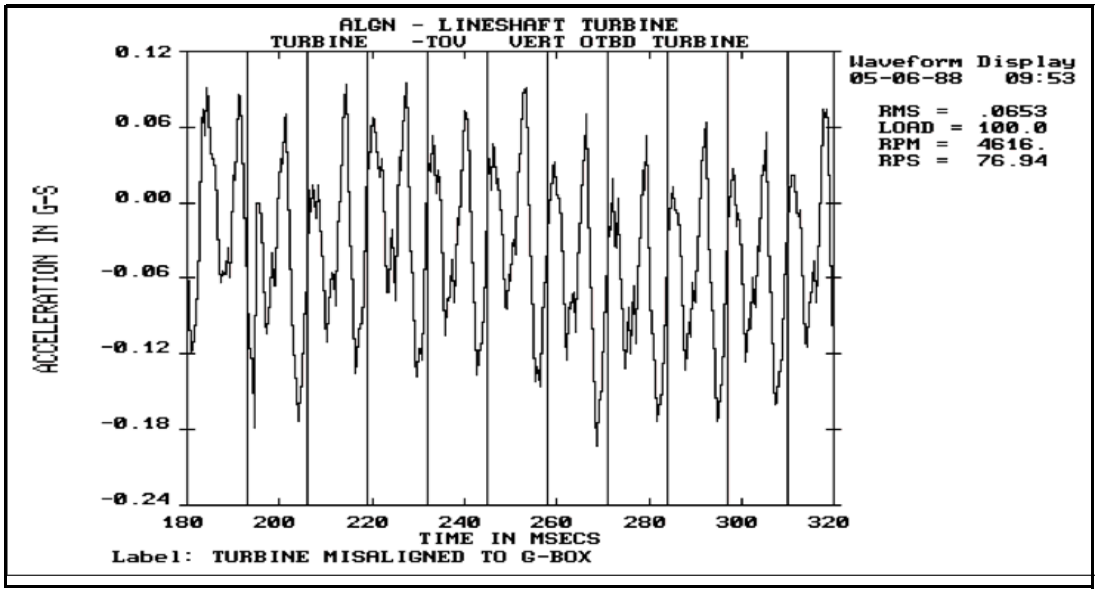


Figure 10

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

List of Spectral Peaks

Machine: (Algn) Lineshaft Turbine
Meas. Point: Turbine - TOV - Vert OTBD Turbine
Date/Time: 05-06-88 / 09:53:04
Amplitude Units: In / Sec Pk
Data Label: Turbine Misaligned to G-Box

Peak No.	Frequency (Hz)	Peak Value	Order Value	Peak No.	Frequency (Hz)	Peak Value	Order Value
1	14.73	.0169	.19	**13**	329.26	.0021	4.28
2	29.61	.0146	.38	**14**	359.18	.0028	4.67
3	40.98	.0028	.53	**15**	384.84	.0110	5.00
4	43.99	.0028	.57	**16**	461.81	.0044	6.00
5	76.59	.0175	1.00	**17**	538.78	.0044	7.00
6	91.55	.0029	1.19	**18**	607.07	.0019	7.89
7	118.25	.0031	1.54	**19**	615.77	.0061	8.00
8	153.93	.0815	2.00	**20**	624.09	.0027	8.11
9	178.91	.0019	2.33	**21**	688.34	.0023	8.95
10	230.91	.0182	3.00	**22**	693.30	.0047	9.01
11	307.92	.0088	4.00	**23**	769.51	.0023	10.00
12	316.68	.0020	4.12	**24**	923.74	.0084	12.01
Total Mag .3270		Subsynchronous .1184 / 13%		Synchronous .3010 / 85%		Nonsynchronous .0480 / 2%	

Table 1

Data Analysis:

1) Dominant synchronous energy (Table 1)
2) Significant 2xTS peak in horizontal position (Figures 7 and 9)
3) 2x almost 5 times the amplitude of 1x (Figure 9)
4) 2 events per shaft revolution (Figure 10)
5) 3x indicates probable coupling damage (Figure 7)

Diagnosis: Offset Misalignment

Corrective Action: Align turbine shaft to gearbox

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The waveform below in Figure 11 shows two clear peaks for each revolution of the shaft. The vertical lines denote one revolution of the shaft. The low G levels indicate little or no impacting.

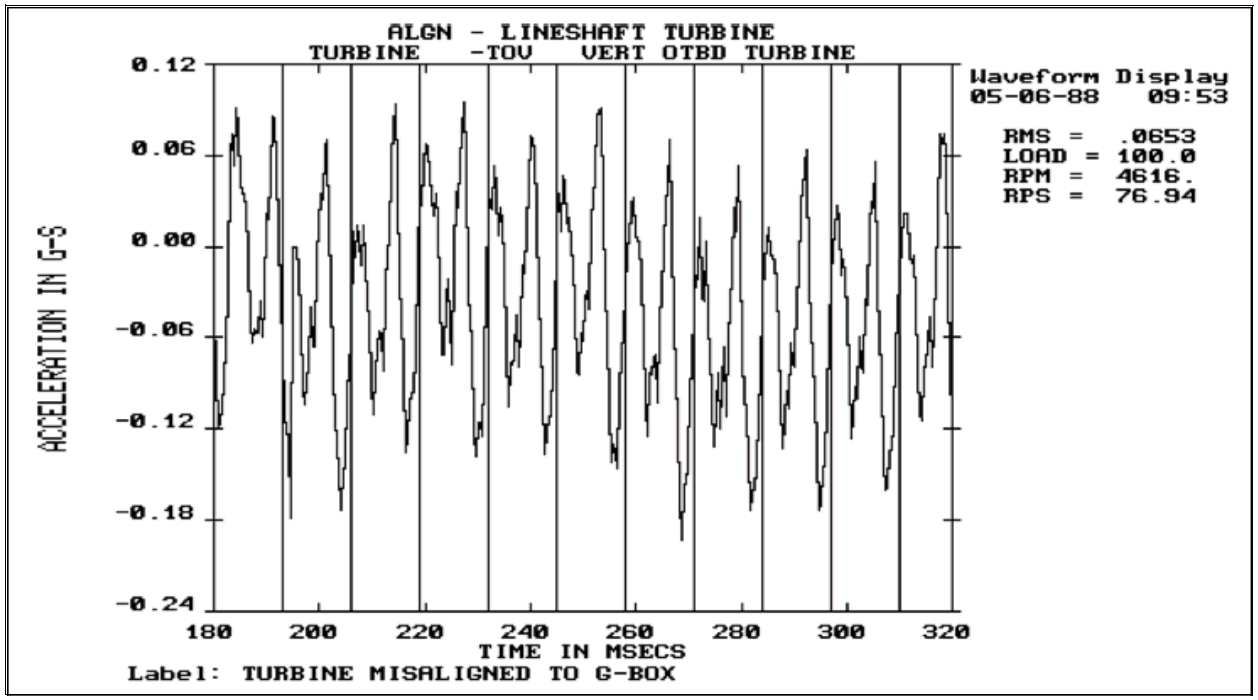


Figure 11

 Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #2 - Axial Piston Pump

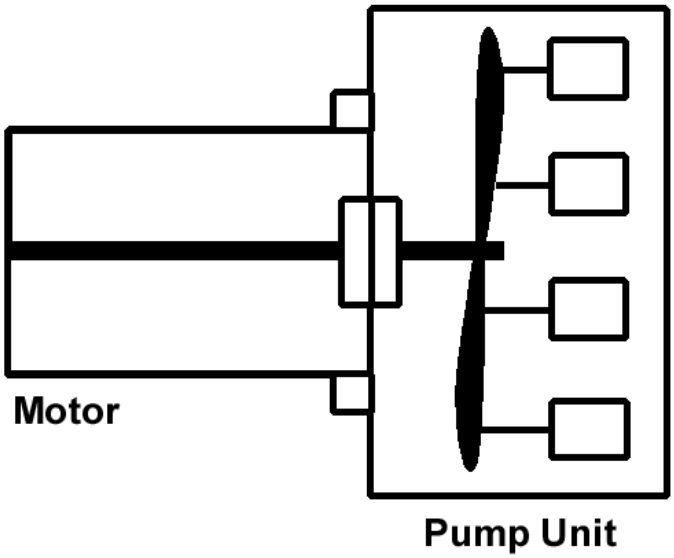


Figure 12

Equipment Data:

Motor and pump are integrated via a “C” face mount. This makes alignment corrections difficult, at best.
There are 9 pistons in the pump.
Each piston is actuated once per revolution of the shaft.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The data from the motor measurement points are displayed in Figure 13. The amplitude of the 1x peak in the outboard axial direction and the inboard radial directions are almost equal. An amplitude of 0.56 in/sec is considered unusually high for an axial reading. The vane pass frequency is responsible for the 9xTS peaks.

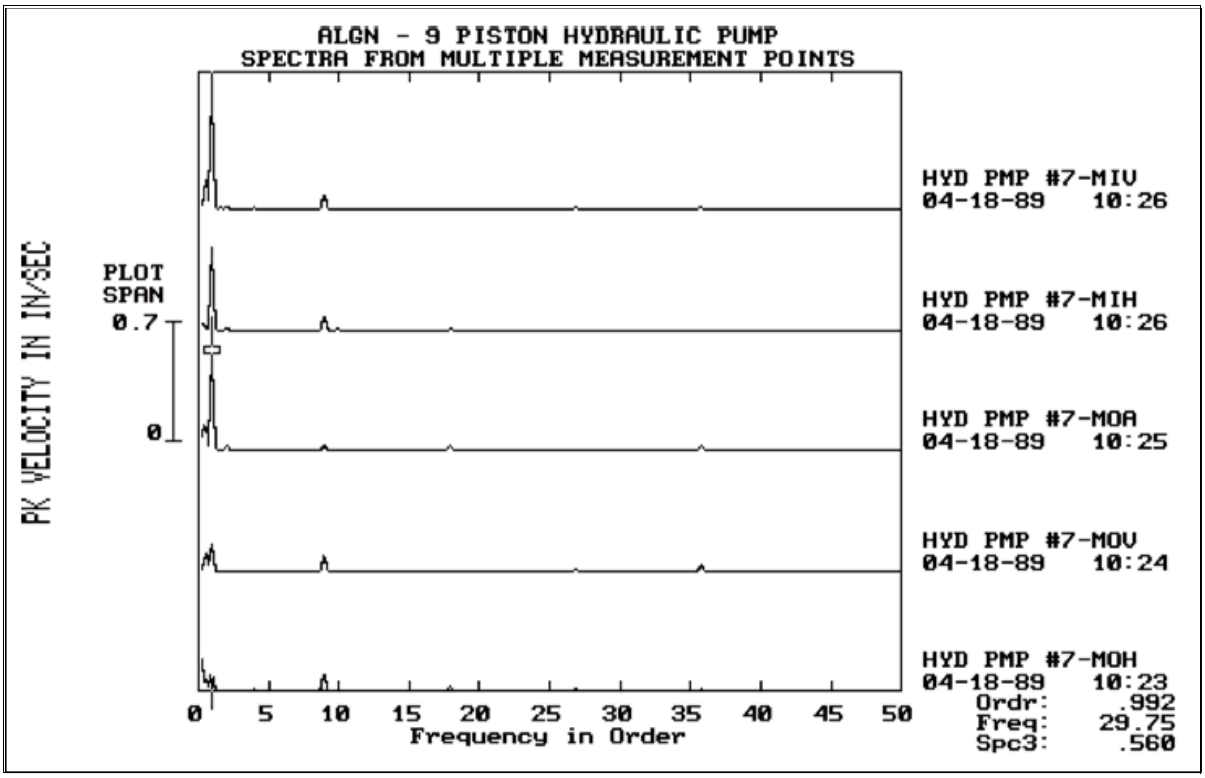


Figure 13

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The pump data is displayed in Figure 14. Measurement point POA is exhibiting an exceptionally high turning speed peak amplitude of over 2 in/sec. The radial points are low by comparison. Without the axial data, accurate analysis of the equipment would be difficult. This machine could easily have been diagnosed as having only an imbalance problem.

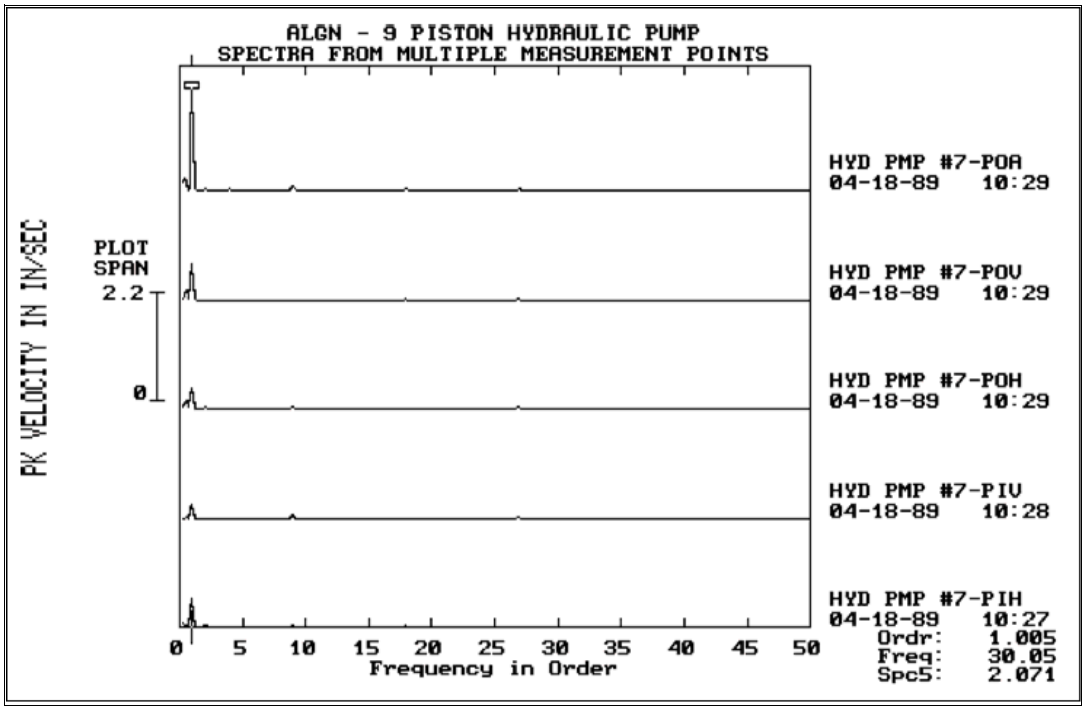


Figure 14

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The single spectrum of the POA measurement point is displayed in Figure 15. The cursor marks the turning speed peak (1 order). A turning speed amplitude of 2 in/sec almost always indicates a severe condition. The overall energy in this spectrum is calculated as 2.1 in/sec. Turning speed is contributing 99% to that.

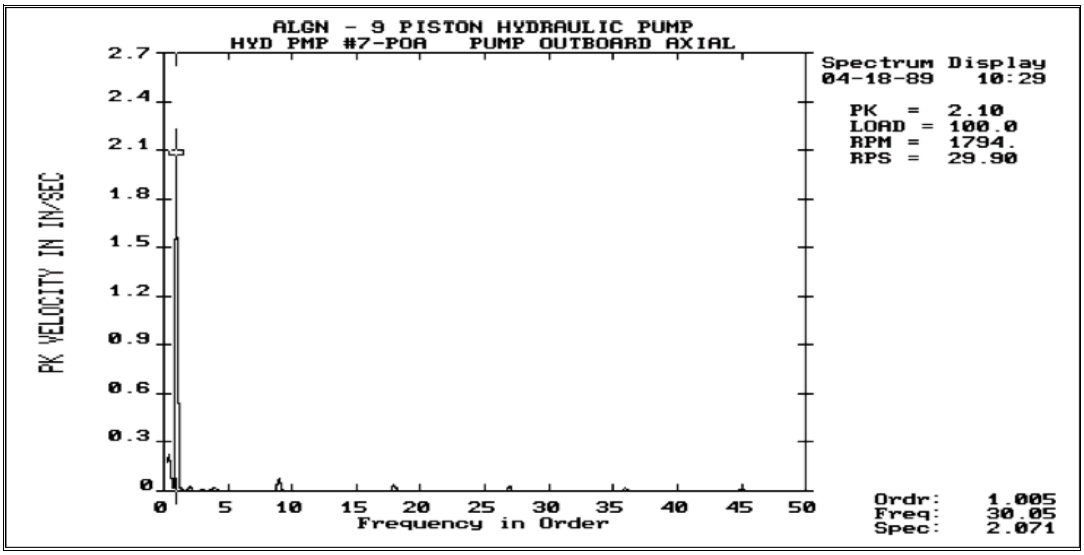


Figure 15

A velocity waveform is useful for confirming relatively low frequency events such as 1x turning speed, etc. The rotating frequency is clearly visible. Harmonic markers help illustrate the sinusoidal shape of the waveform, and confirm that most of the energy is coming from the turning speed of the shaft. Since the waveform is very sinusoidal, the peak amplitudes are close to those in the spectrum.

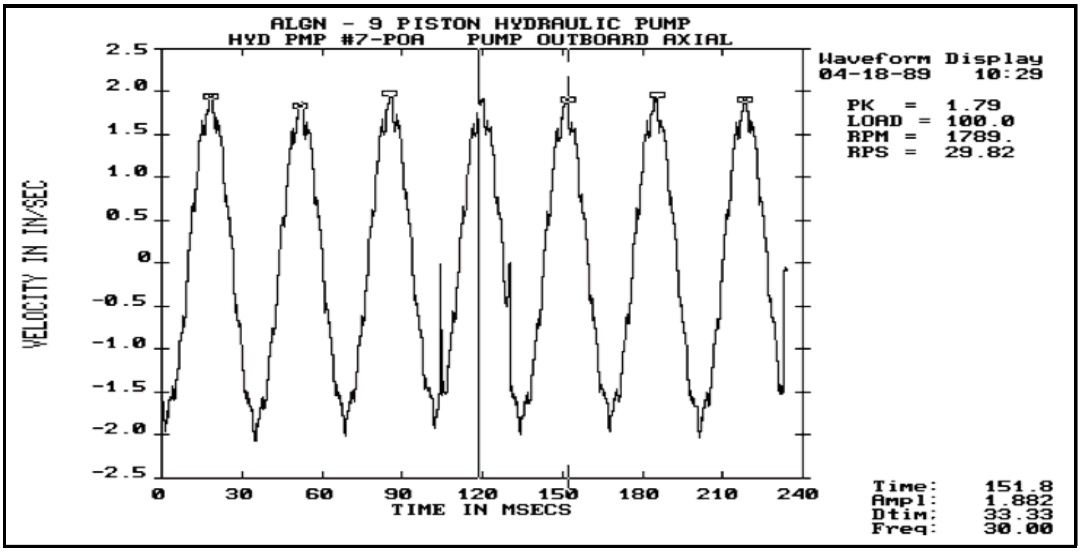


Figure 16

 Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Machine: (ALGN) 9 PISTON HYDRAULIC PUMP
Meas. Point: HYD PMP #7-POA --> PUMP OUTBOARD AXIAL
Date/Time: 18-Apr-89 10:29:38 RPM= 1800. Units=In/Sec PK

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	10.14	.2412	.338	13	447.73	.0021	14.924
2	30.05	2.0670	1.002	14	508.13	.0040	16.937
3	59.59	.0254	1.986	15	537.45	.0378	17.915
4	90.51	.0089	3.017	16	567.81	.0027	18.927
5	118.77	.0226	3.959	17	597.55	.0022	19.918
6	137.20	.0025	4.573	18	774.86	.0035	25.828
7	148.93	.0043	4.964	19	804.55	.0317	26.818
8	178.62	.0033	5.954	20	834.69	.0027	27.823
9	209.34	.0022	6.978	21	1044.56	.0022	34.818
10	238.30	.0025	7.943	22	1073.42	.0163	35.780
11	268.39	.0895	8.946	23	1340.95	.0111	44.698
12	298.02	.0041	9.934	24	1371.24	.0020	45.707

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
2.0839	.2412 / 1%	2.0671 / 98%	.1073 / 0%
(ELECTRICAL=	0.00E+00/ 0%	FLOOR= .0167	SPEC CREST= 1.30)

Table 2

Data Analysis:

1) Integrally mounted machine (Figure 12)
2) Mostly synchronous energy (Table 2)
3) Dominant 1x peak in the axial direction (Figures 13,14, and 15)
4) Little radial energy in pump (Figure 14)
5) Periodic, sinusoidal waveform (Figure 16)
6) Little impacting (Figure 16)

Diagnosis: Angular misalignment

Corrective Action: Align pump to motor

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #3 - Centrifugal Air Compressor

A large guard protected the coupling on this unit. Most of the vibration was occurring at 2xTS. Therefore, since coupling damage is usually manifest at 3xTS, it was probably in good condition. The data indicates some looseness in the machine.

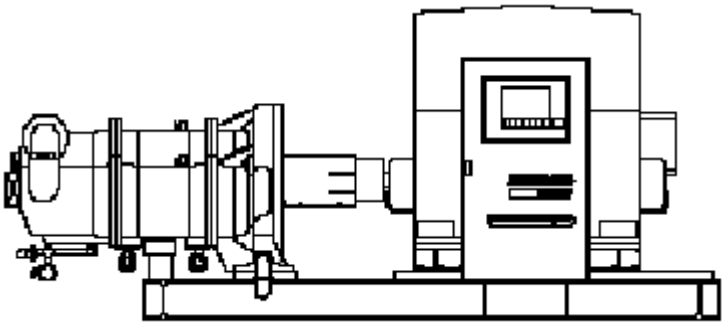


Figure 17

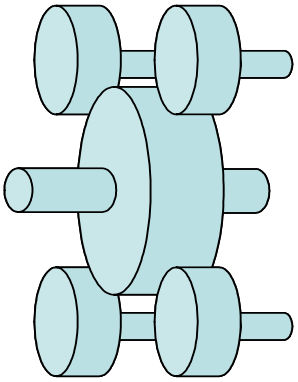


Figure 18

*Another type of Centrifugal Air Compressor

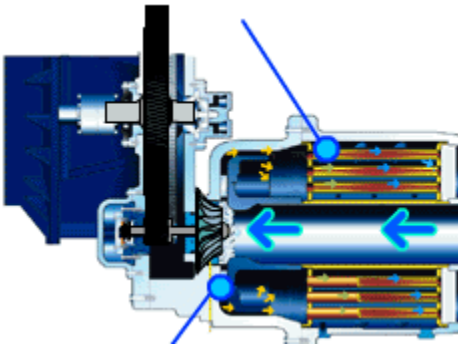


Figure 19

Equipment Data: The motor drives the main (Bull Gear). The Bull Gear meshes with the Pinion gears. Since the Pinion gears have fewer teeth than the Bull Gear, their shafts are rotating at a much higher frequency than the Bull Gear. In some cases, all the Pinions may not have the same number of teeth. In those cases, there will be multiple shaft turning speeds in the gearbox.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The 2xTS peak is the dominant frequency on all the motor radial measurement points. The amplitude is highest on the inboard vertical positions. 1xTS is approximately 0.008 in/sec. 2xTS is in excess of 0.18 in/sec. This ratio of 22:1 is definitely a cause for concern.

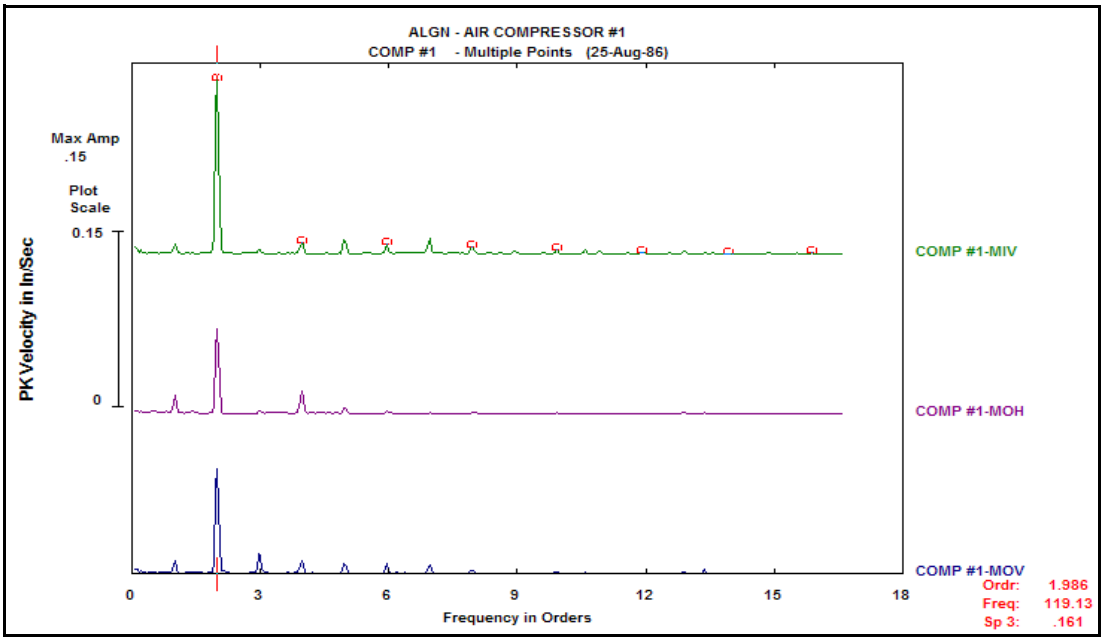


Figure 20

Machine: (ALGN) AIR COMPRESSOR #1
Meas. Point: COMP #1 -MIV --> MOTOR INBOARD VERTICAL
Date/Time: 25-Aug-86 09:12:55 RPM= 3600. Units=In/Sec PK
Data Label: Priority: 0

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	12.50	.0033	.208	13	381.69	.0017	6.361
2	17.50	.0015	.292	14	391.32	.0011	6.522
3	23.25	.0013	.387	15	398.30	.0011	6.638
4	46.23	.0017	.770	16	416.75	.0141	6.946
5	60.36	.0087	1.006	17	476.47	.0067	7.941
6	119.13	.1609	1.986	18	516.14	.0020	8.602
7	133.48	.0016	2.225	19	536.18	.0032	8.936
8	166.17	.0017	2.769	20	595.53	.0034	9.926
9	178.87	.0039	2.981	21	635.31	.0034	10.589
10	238.35	.0107	3.972	22	655.20	.0024	10.920
11	298.08	.0120	4.968	23	774.18	.0025	12.903
12	357.32	.0082	5.955	24	893.31	.0015	14.889

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.1632	.0042 / 0%	.1612 / 98%	.0253 / 2%
(ELECTRICAL= 0.00E+00/ 0%		FLOOR= .0079	SPEC CREST= 1.14)

Table 3

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Data Analysis:

1) High 2xTS peak in the radial direction (Figure 19)
2) Greater than 2:1 ratio of 1xTS to 1xTS (Figure 19)
3) Dominant synchronous energy (Table 3)
4) Low amplitude 1xTS amplitude (Figure 19)

Diagnosis: Offset Misalignment

Corrective Action: Align the motor shaft to the input shaft on the gearbox.

 Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #4 - Turbine Generator

Figure 21

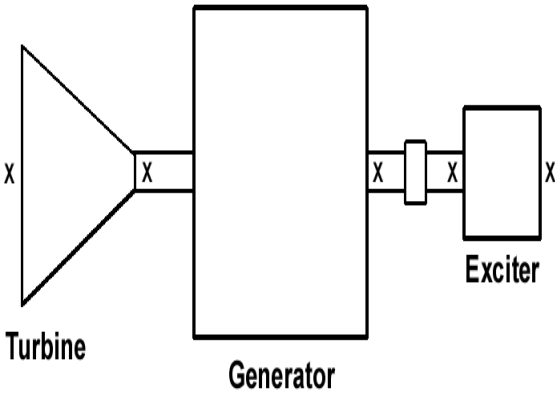


Figure 22

Equipment Data:
5 Mega Watt Generator

Equipment Notes:
The turbine and generator are on a common shaft. Therefore, there is no coupling between the two. The turbine / generator combination outweigh the exciter by a factor of approximately 25. The exciter is coupled to the outboard end of the generator. A lot of energy is present on the exciter end of the machine. Exciter looseness is suspected due to the weight distribution. Either condition could generate the elevated 3xTS peak, or the coupling could be dry or locked.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The turbine data seems to be within acceptable operating parameters. However, comparing the generator points to the exciter points, the exciter points are exhibiting higher amplitudes in the 1xTS to 8xTS frequency range. (Figure 23) It may be useful to examine the time waveform to help determine the main cause of this much vibration.

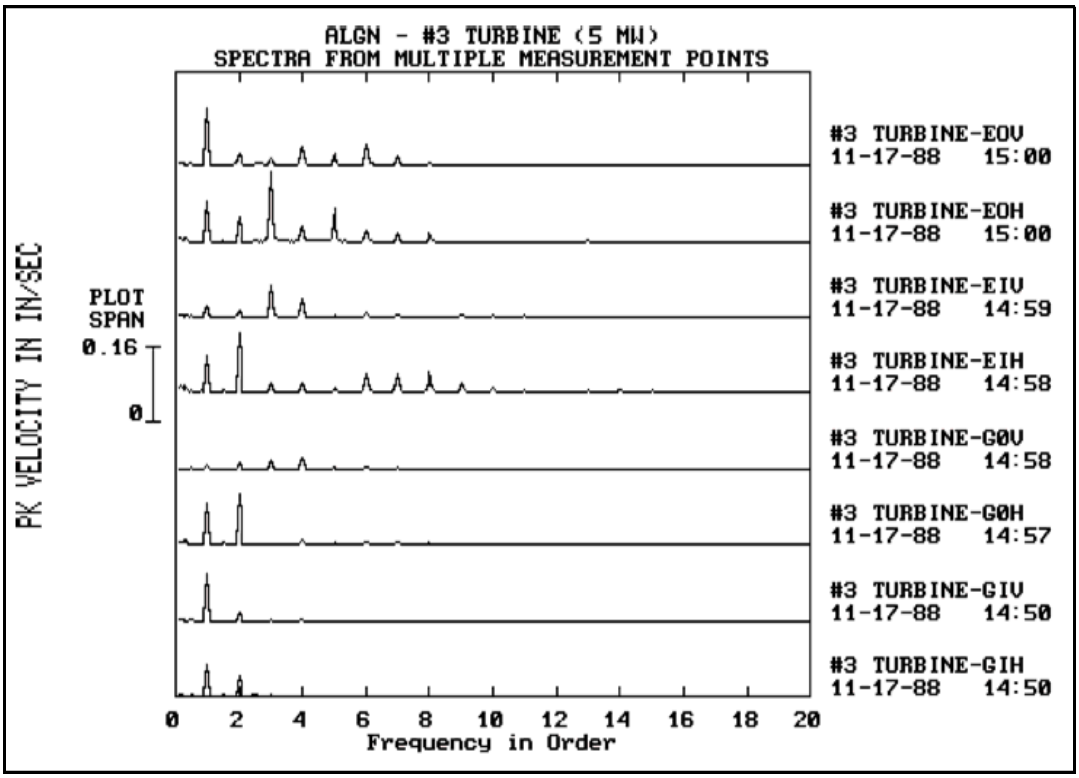


Figure 23

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Exciter measurement points indicate an excessive amount of energy in all directions. Note the relatively high amplitudes of the axial points compared to the radial points. A great deal of the energy is coming from the range of 1xTS to 4xTS. Some turning speed harmonics in the radial direction are normal in sleeve bearing machines. However, further analysis is needed to resolve the root cause of the high axial energy. (Figure 24)

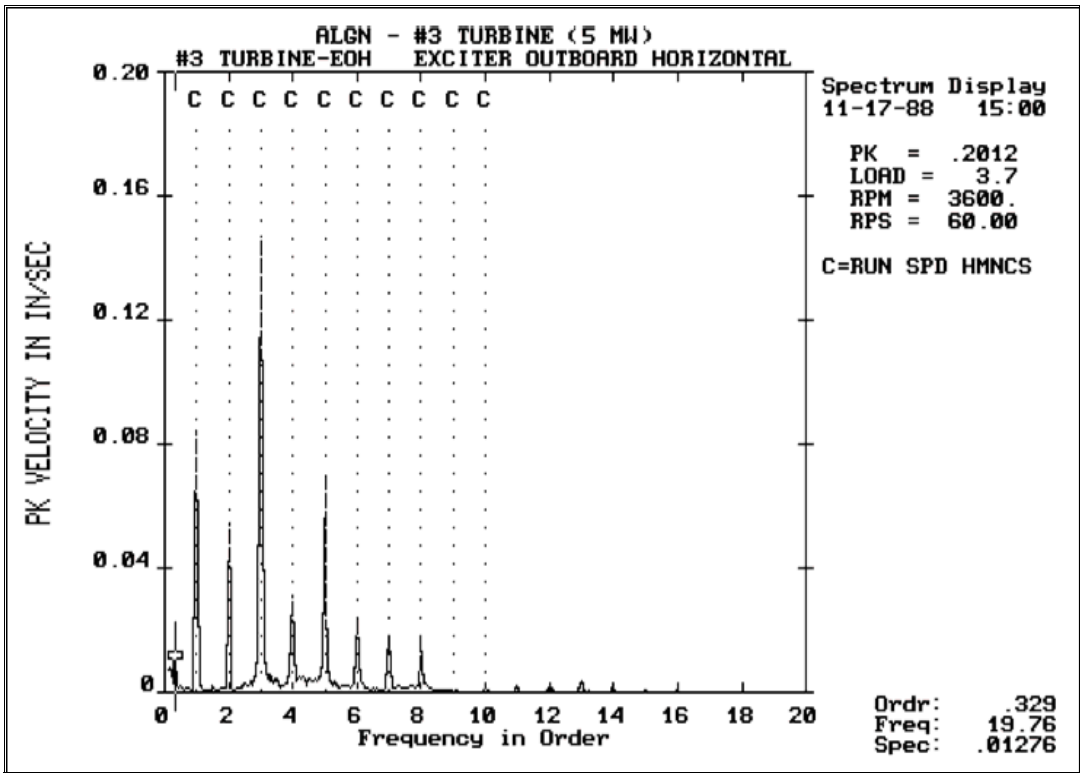


Figure 24

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The single spectrum plot from EOH is exhibiting turning speed harmonics with an elevated 3xTS peak. Turning speed harmonics themselves are not necessarily a cause for concern. The high 3xTS peak does deserve scrutiny. There appears to be somewhat of a raised noise floor between 2xTS and 8xTS. This may indicate a degree of looseness. The 2xTS peak is greater than half the amplitude of the 2xTS peak. (Figure 25)

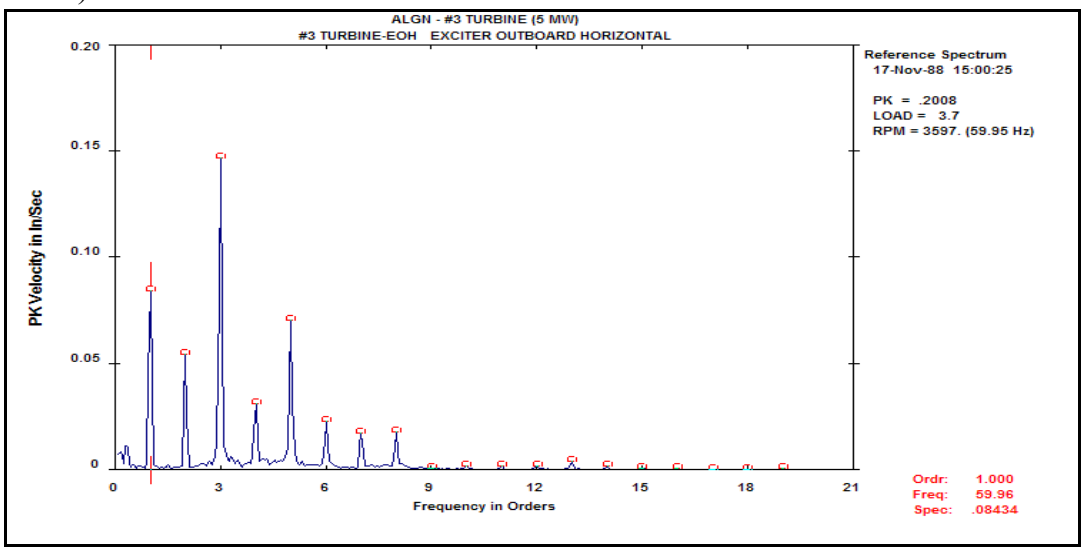


Figure 25

When the exciter waveform is examined, misalignment is evident by the periodic, repetitive pattern. Had looseness been more severe a less repetitive pattern would be expected. (Figure 26)

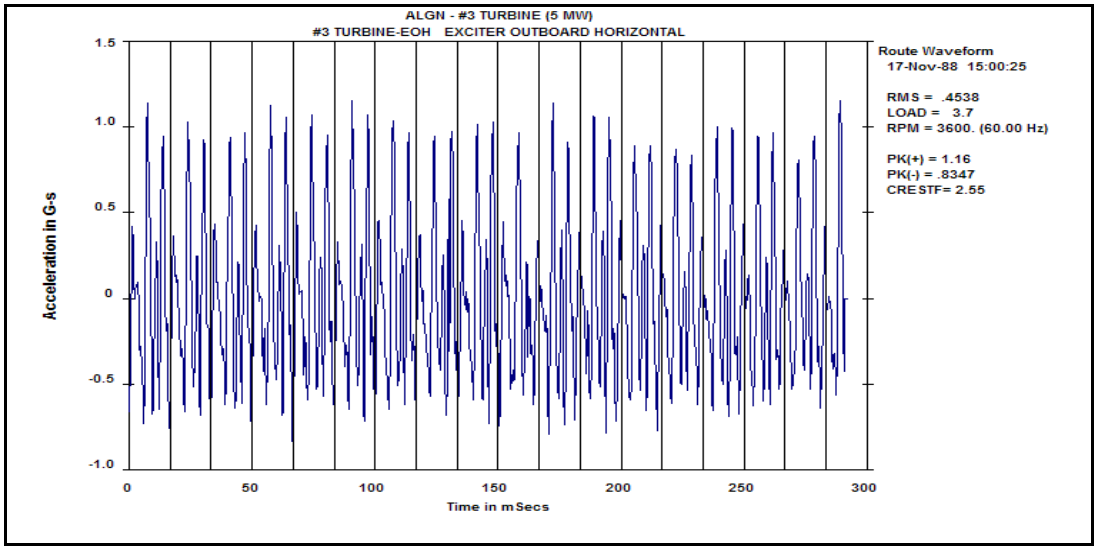


Figure 26

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Examination of the spectrum in the axial direction shows that the synchronous energy is prevalent. Of particular interest is the 1xTS to 4xTS range. The 1xTS to 2xTS amplitude ratio is still present. Although the 3xTS peak is lower in amplitude in the axial direction than in the radial, it is still significant because of its' broad "skirt". This tends to indicate imminent coupling damage. (Figure 27)

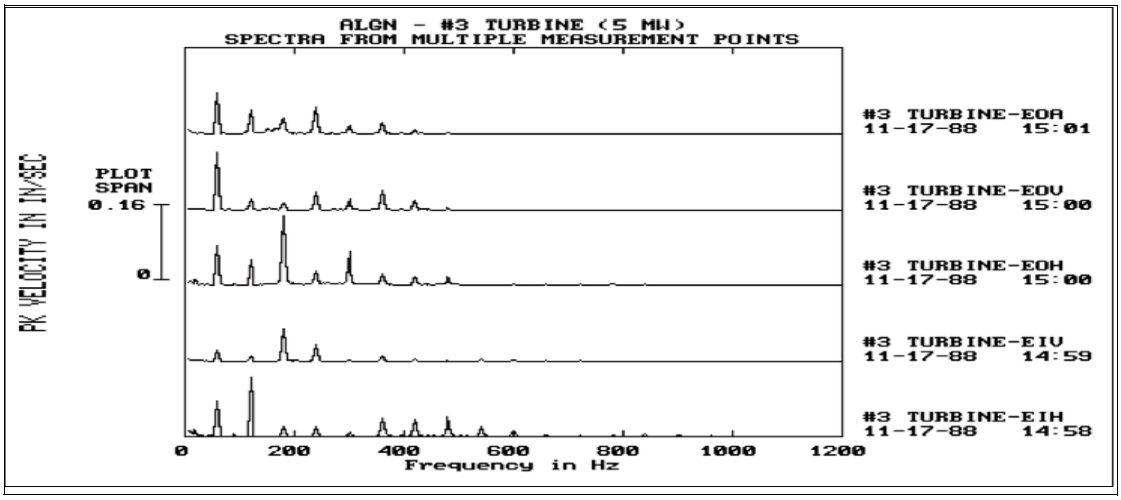


Figure 27

Analysis of the exciter outboard axial waveform indicates a very simple, cyclic, somewhat sinusoidal pattern. Three or four peaks per shaft revolution are present. This evidence indicates that repetitive force is driving the vibration. Amplitudes are relatively low at plus or minus 1g. (Figure 28)

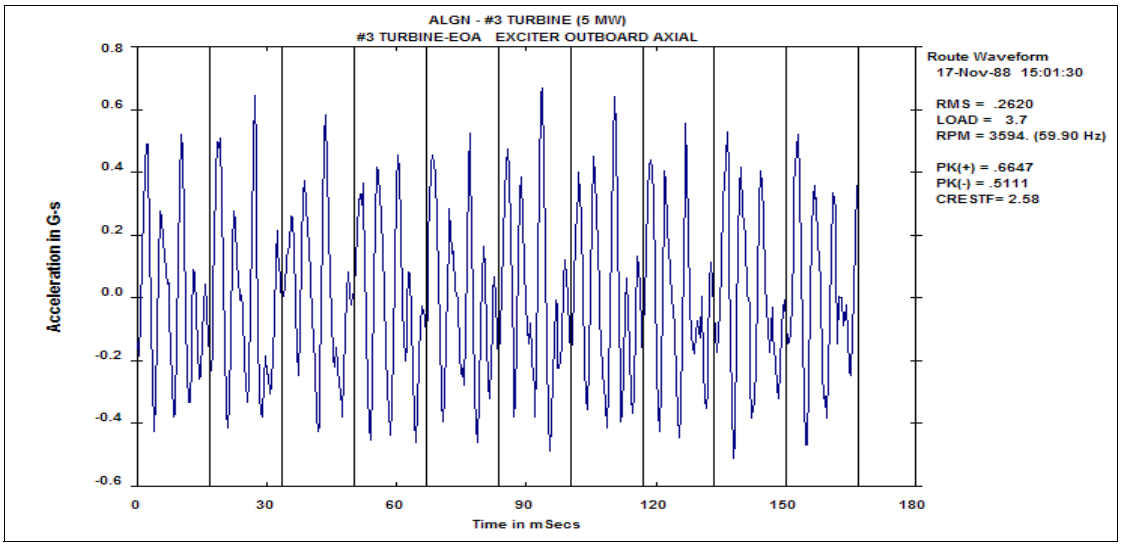


Figure 28

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

LIST OF SPECTRAL PEAKS
****

Machine: (ALGN) #3 TURBINE (5 MW)
Meas. Point: #3 TURBINE-EOA --> EXCITER OUTBOARD AXIAL
Date/Time: 17-Nov-88 15:01:30 RPM= 3594. Units=In/Sec PK

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	16.28	.0037	.272	13	239.44	.0591	3.997
2	31.13	.0022	.520	14	268.57	.0019	4.484
3	59.94	.0854	1.001	15	289.41	.0025	4.832
4	119.67	.0527	1.998	16	300.19	.0195	5.012
5	131.01	.0028	2.187	17	310.39	.0035	5.182
6	151.42	.0123	2.528	18	349.13	.0027	5.829
7	160.26	.0078	2.676	19	359.77	.0248	6.006
8	166.47	.0134	2.779	20	370.34	.0035	6.183
9	178.99	.0368	2.988	21	395.11	.0027	6.596
10	193.52	.0023	3.231	22	419.79	.0096	7.008
11	201.45	.0021	3.363	23	430.64	.0019	7.190
12	210.81	.0034	3.519	24	479.64	.0033	8.008

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.1285	.0043 / 0%	.0934 / 53%	.0218 / 3%
(ELECTRICAL=	.0854 / 44%	FLOOR= .0170	SPEC CREST= 1.57)

Table 4

Data Analysis

1) Prevalent synchronous energy (Figure 28)
2) High 1xTS to 2xTS ratio (Figures 24 and 26)
3) Elevated 3xTS peak (Figure 24)
4) Repetitive, periodic waveform (Figures 25 and 27)
5) Low impacting (Figures 25 and 27)

Diagnosis: Misalignment between the generator and exciter

Corrective Action: Align exciter to generator; modify exciter support structure

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #5 - Upper Quench Fan

Unbalance exhibits an elevated 1x peak usually in the radial direction, and could have been suspected on this machine. However, the data in Figure 30 was collected in the axial direction. Angular misalignment is manifest as an elevated 1x peak in the axial direction.

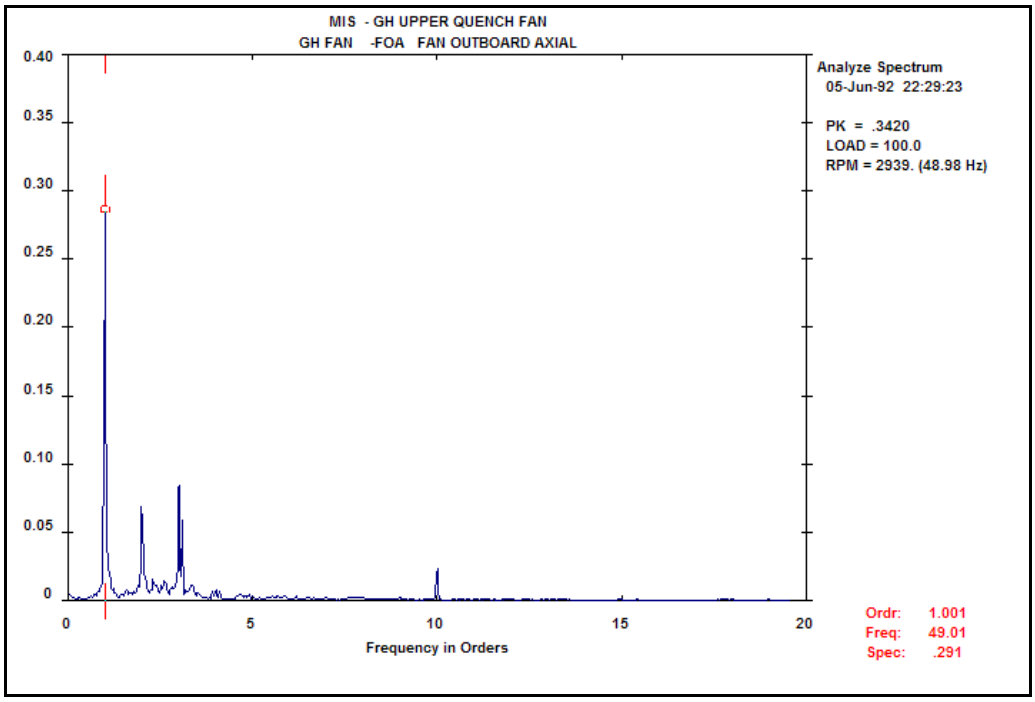


Figure 29

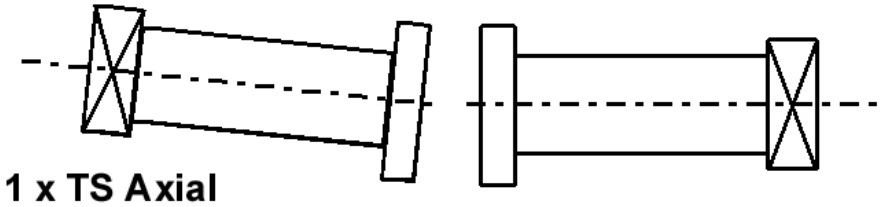


Figure 30

With angular misalignment the shafts are being forced away from each other.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The time waveform pattern in Figure 31 is also characteristic of a machine with misalignment. Note the W or M appearance per revolution.

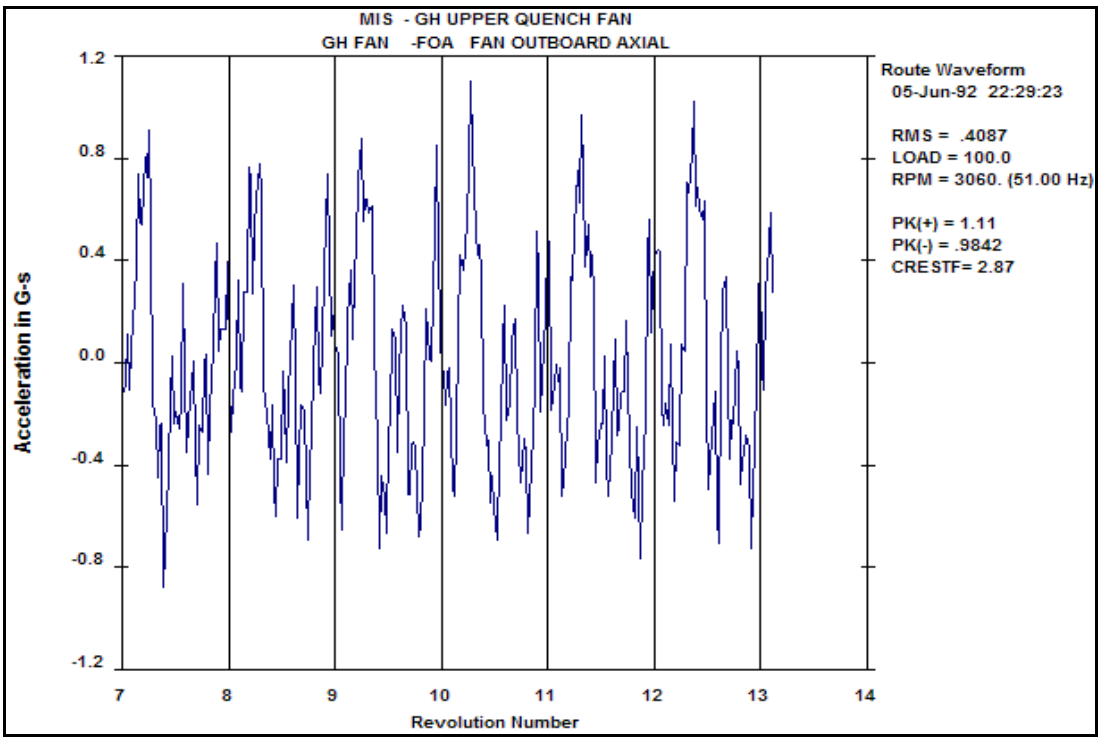


Figure 31

Use the Time Waveform patterns to help verify a problem.

If Phase data was taken on this machine it would show a 180 degrees phase shift across the coupling.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The post alignment spectral data is shown in Figure 32. The vibration was reduced by approximately 50%. The presence of turning speed harmonics now indicates some residual looseness.

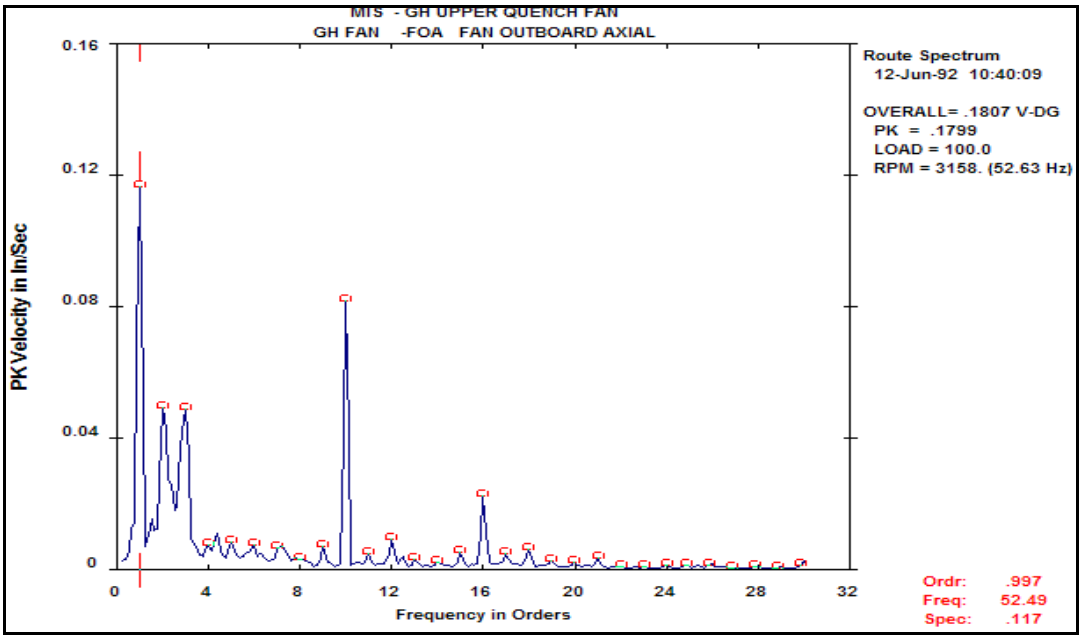


Figure 32

The corrective measure has also affected the time domain data. Looseness is usually suspected with a pattern as seen in Figure 33.

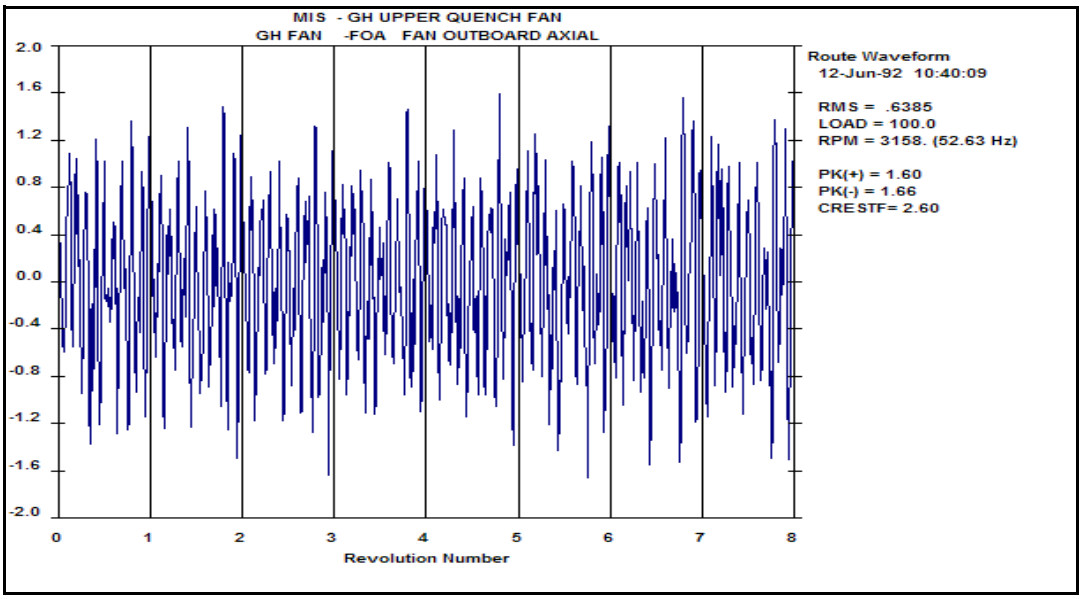


Figure 33

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

 Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Mechanical Looseness

Section 4

Objectives

- Define Looseness
- Describe the categories
- Determine some cause(s)
- Identify the characteristic(s)
- Determine corrective action(s)

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Mechanical Looseness

Defined: The state or condition of a rotating element where its fasteners or rotating element are no longer held fast or rigid to its host.

The looseness in a machine may be classified by one of two categories.

Structural Looseness Includes:

- Base Mount
- Split Casings
- Bearing Caps
- Bearing Supports

Rotating Element Looseness Includes:

- Impellers
- Fans
- Bearings
- Couplings

Fasteners, hold down bolts, etc. may work loose due to excessive or inherent vibration causing structural looseness. Also, grouting and other supporting mechanisms can be compromised over time causing a looseness condition.

Other characteristics of looseness:

- Tends to be directional in nature
- Some machines will have inherent looseness

Waveform:

- Non-periodic
- Non-sinusoidal

Spectrum

- Turning speed harmonics
- Occasionally fractional harmonics may appear
- Broad based “skirts” on spectral peaks
- Possible elevated noise floor
- Many harmonics of turning speed

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #1 - Pump Motor with Soft Foot

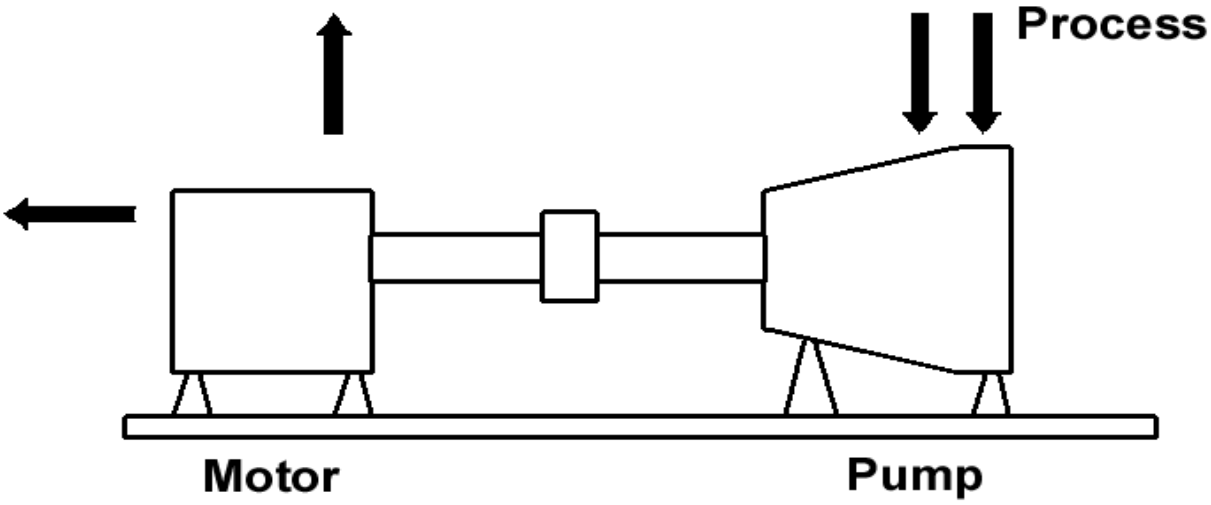


Figure 1

Equipment Data:

- Pump supplies fiberglass mat process
- MOA and MIV show highest vibration levels
- MOA spectrum suggests possible misalignment
- View waveform in G's to confirm looseness.
- A shim has vibrated out from under inboard foot of motor and into the grease surrounding it. It was replaced while the machine was running with no down time.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Turning speed harmonics out to 6xTS are significant. The Motor Inboard Vertical point appears to have the highest level of vibration. (See Figure 2)

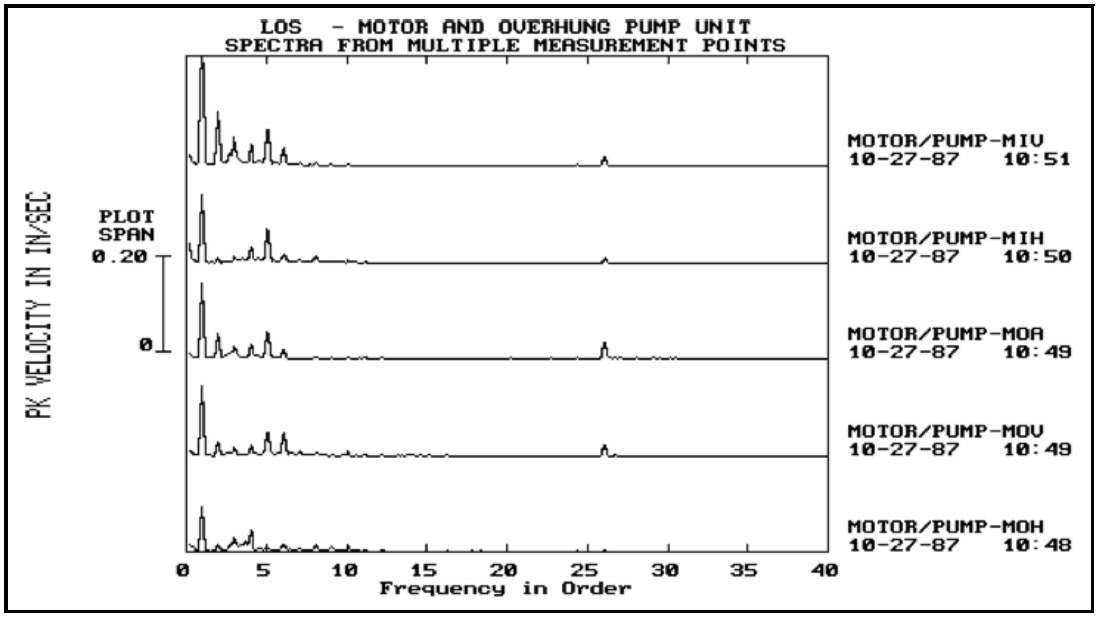


Figure 2

Figure 3 is an expanded view of Figure 2. MIV shows the highest amplitude peaks. Note the relative amplitudes of the five measurement points. Often, this comparison will help determine the type of looseness that is in the machine.

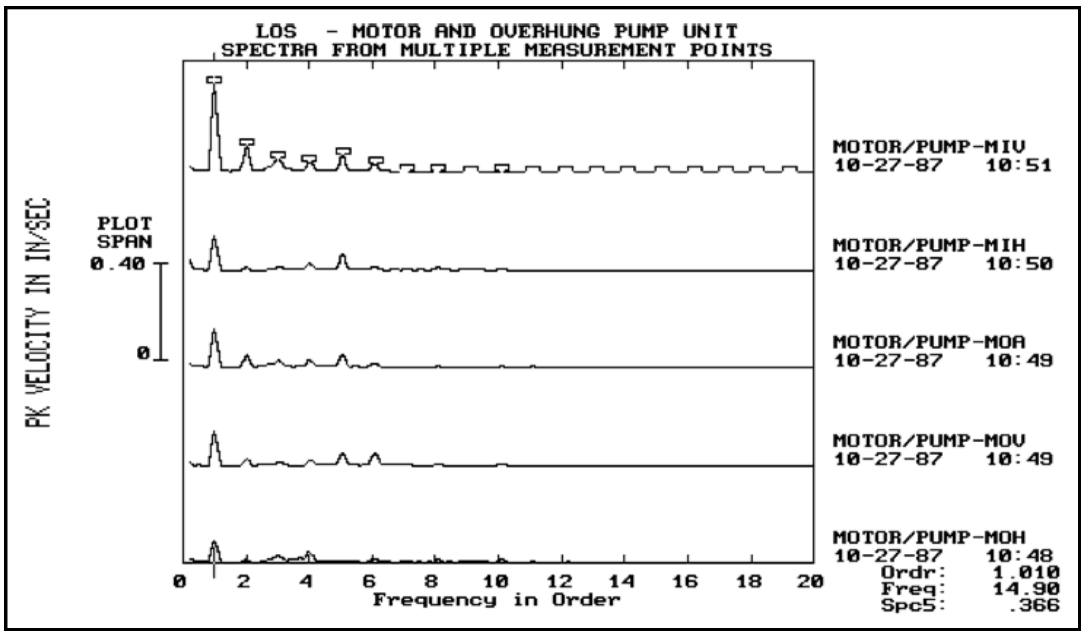


Figure 3

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The spectrum in Figure 4 shows a 1xTS peak with a mound of energy between 3xTS and 5xTS. The overall amplitude remains relatively low at 0.1275 IPS, with harmonics extending to 10xTS.

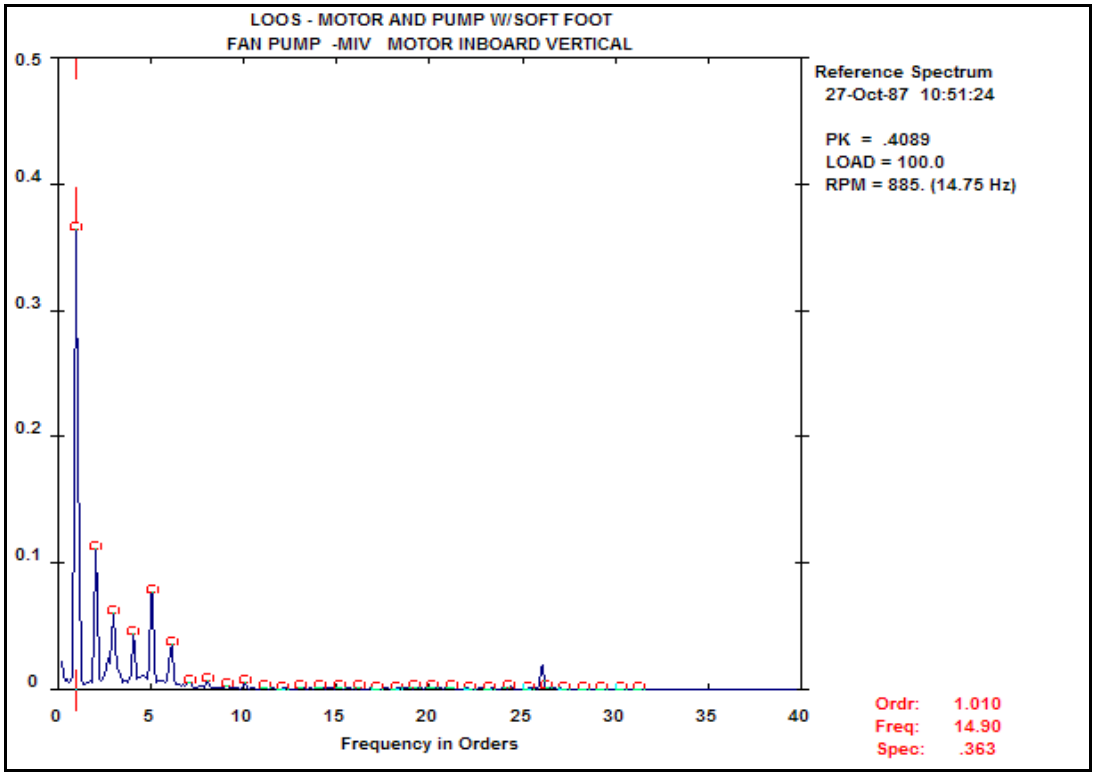


Figure 4

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The time domain is exhibiting a classic pattern of looseness. There is no real repetition in the normalizing frequency. Also, note the non-sinusoidal, random impacting pattern. (See Figure 5)

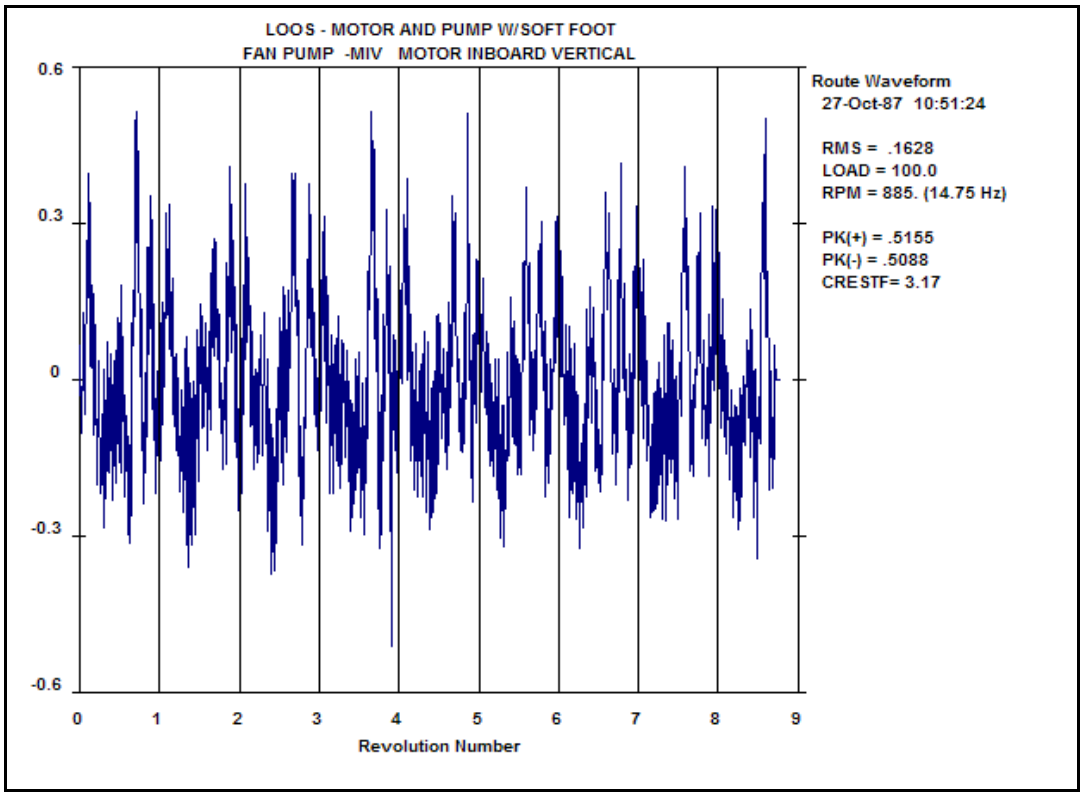


Figure 5

Data analysis

- Elevated turning speed harmonics
- Inboard, vertical location
- Non-sinusoidal, non-periodic waveform with random impacting

Diagnosis

Structural looseness, possibly in the base mounts.

Corrective Action

Check for soft foot, loose fasteners, or cracked grout.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #2 - Torsional Looseness

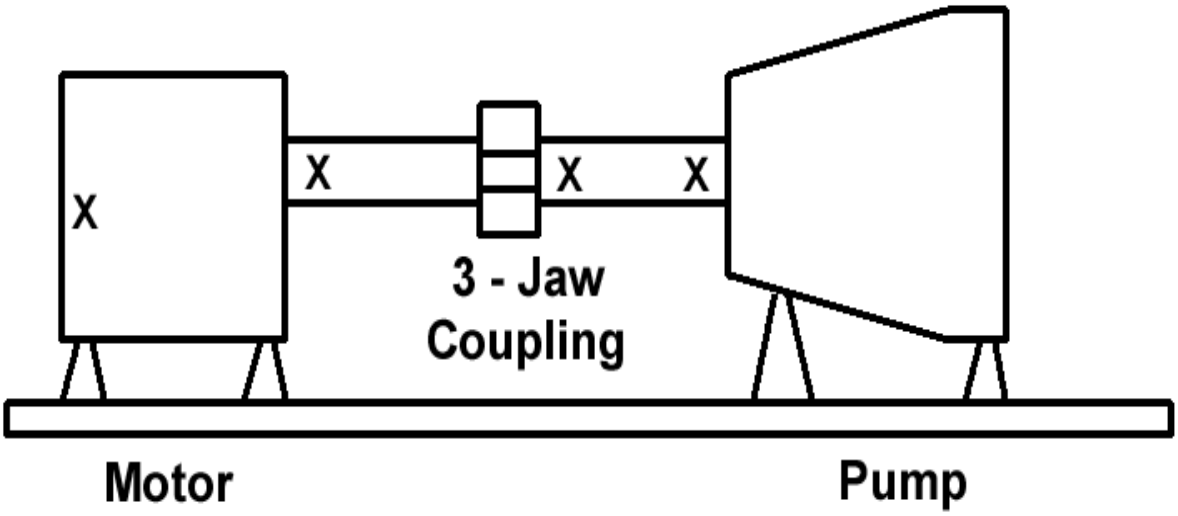


Figure 6

Equipment Information:

- 30 to 40 Horsepower motor
- 3-Jaw coupling
- Overhung pump:
  - * Takes up less space
  - * Less expensive
  - * Pump housing is stationary
  - * Disassembly and repairs are easier

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The data in Figure 7 is from a motor driving a pump through a 3-jaw coupling. The apparent 3XTS peak is dominant on all the pump points.

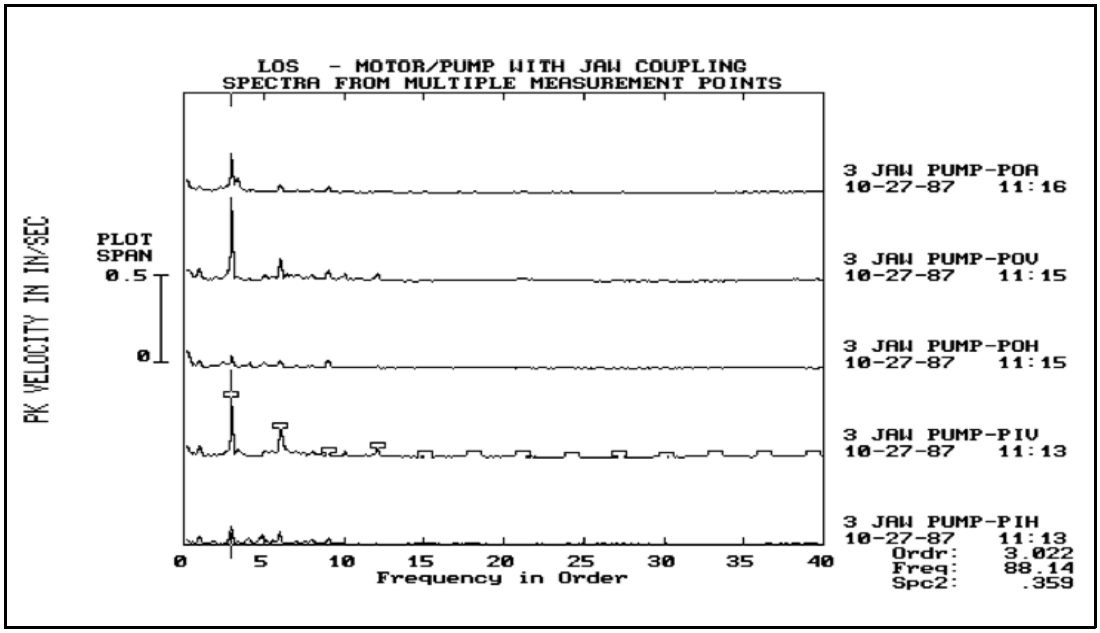


Figure 7

A great deal of energy is attributable to the 3xTS and harmonics. (See Figure 8)

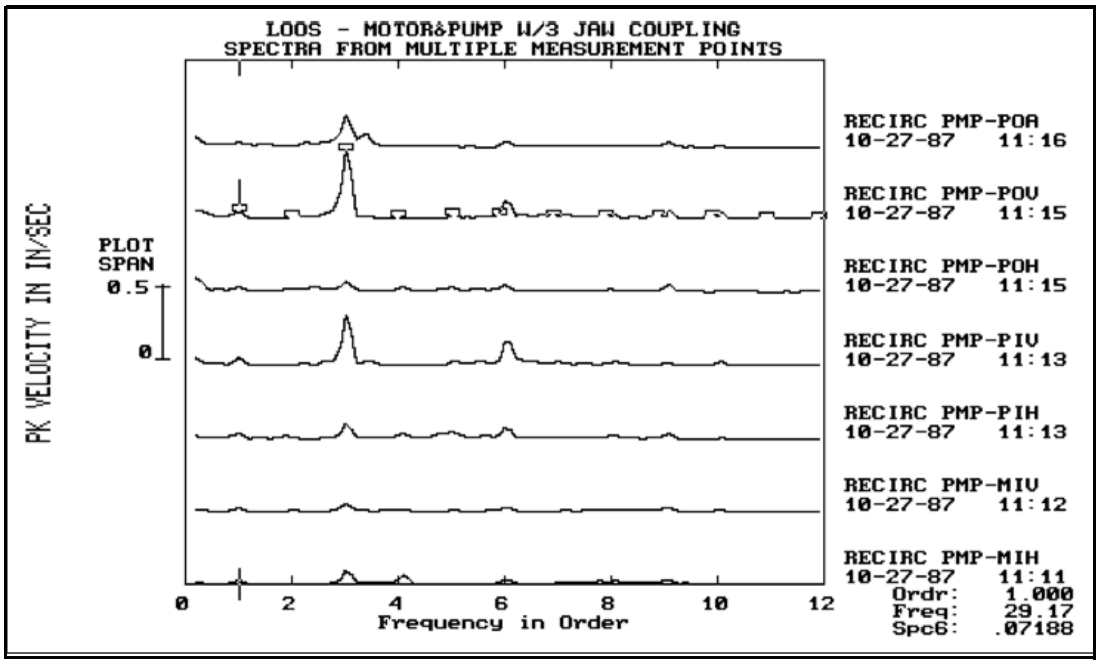


Figure 8

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

With the 3xTS and harmonics out to 9xTS in the POV data in Figure 9, looseness is strongly indicated.

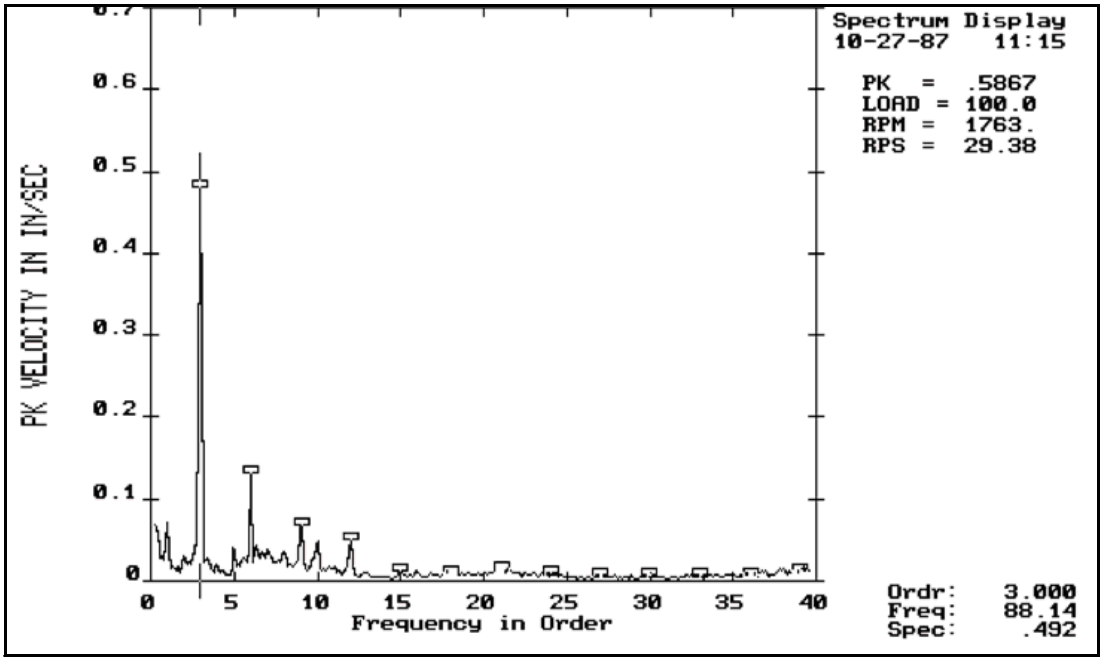


Figure 9

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

A non-periodic, random impacting, non-sinusoidal waveform, with approximately a 12 G swing such as the one in Figure 10 indicates severe looseness.

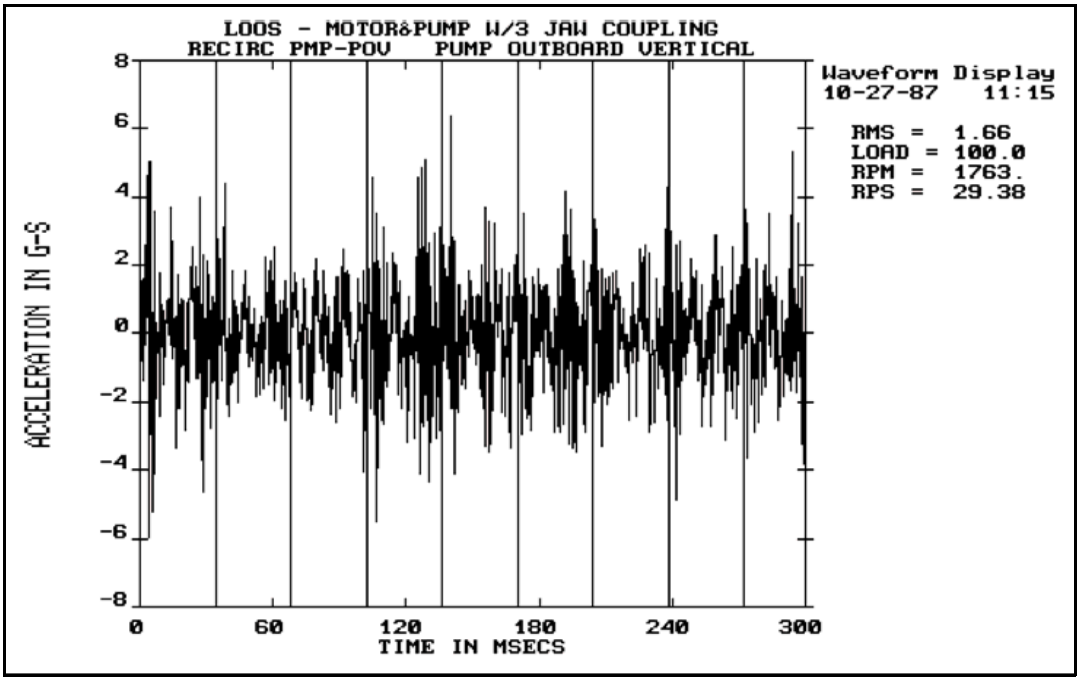


Figure 10

Data Analysis:

- Dominant 3X turning speed with harmonics
- Strongest in the vertical directions
- Non-sinusoidal, non-periodic waveform with random impacting

Diagnosis:

Torsional looseness, probably in the coupling

Corrective Action

Inspect / replace the insert in the coupling
Coupling replacement may be in order

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #3 - Pump Driven by Motor

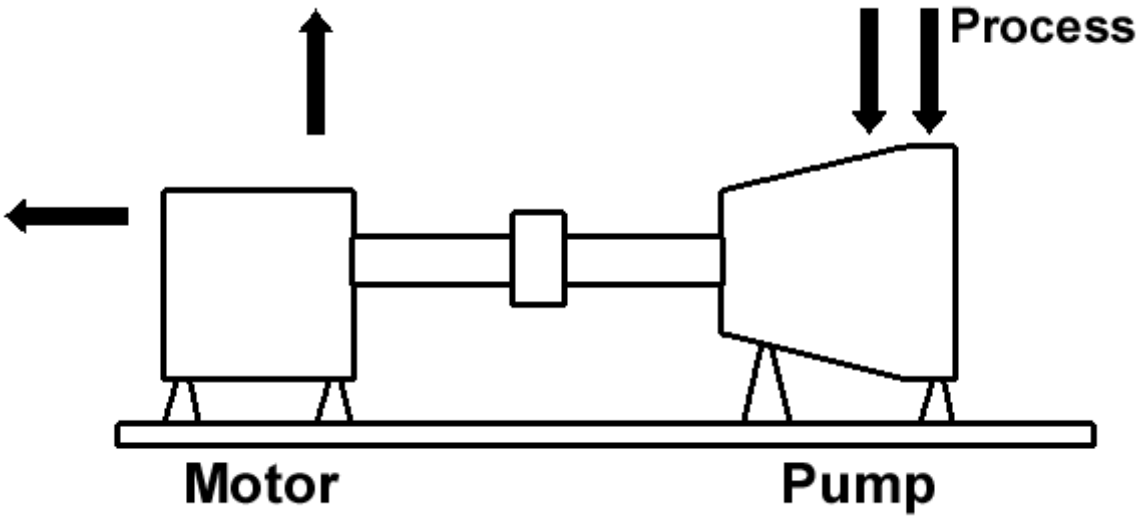


Figure 11

This case history illustrates a looseness fault as it develops over an eight month period. A brief history is given. There is no waveform to analyze.

April	Significant turning speed peak with no appreciative harmonic activity
May	Broad-band energy around elevated 1x and 3xTS peak
June / July	Many harmonics of turning speed now appear
August	Over amplitudes increasing
September	No data collected
October	“Fractional” TS harmonics appear
November	Fractional harmonics still present, scale increases to 0.6 IPS
December	Sharp increase in peak amplitudes; overall exceeds 1.0 IPS

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Elevated 1xTS, but no significant harmonics, indicating a probable unbalance problem. Some looseness is evident by the presence of low amplitude TS harmonics.

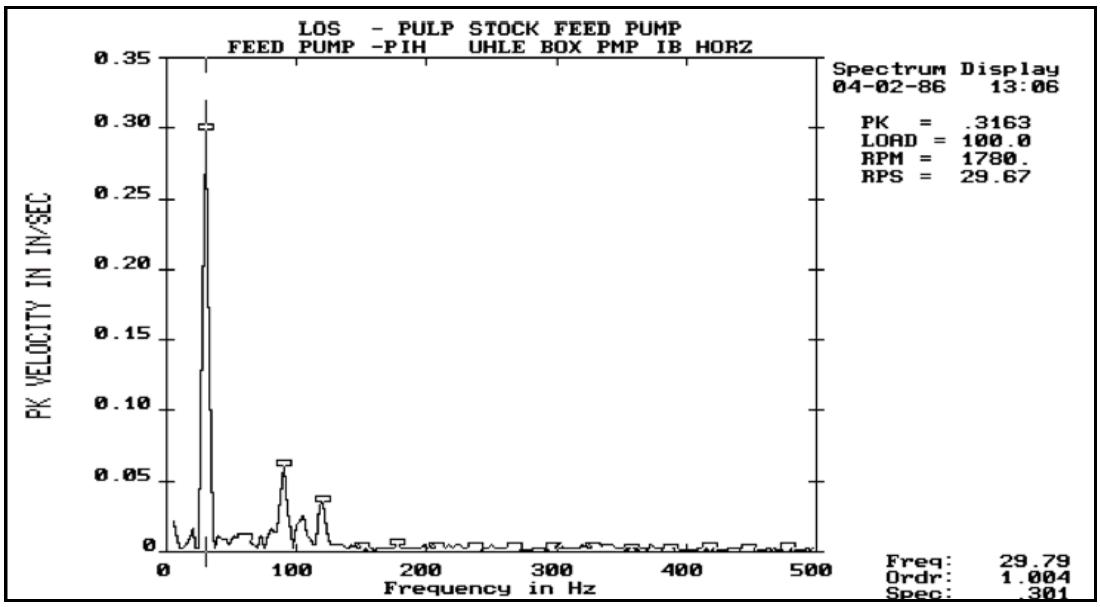


Figure 12

The data from May reveals that the 1X has actually reduced in amplitude. The overalls have also decreased. The second significant peak appears to be 2xTS. Closer inspection reveals that it is actually 3x. The broad skirt around 3x is a good indicator of looseness. (See Figure 13)

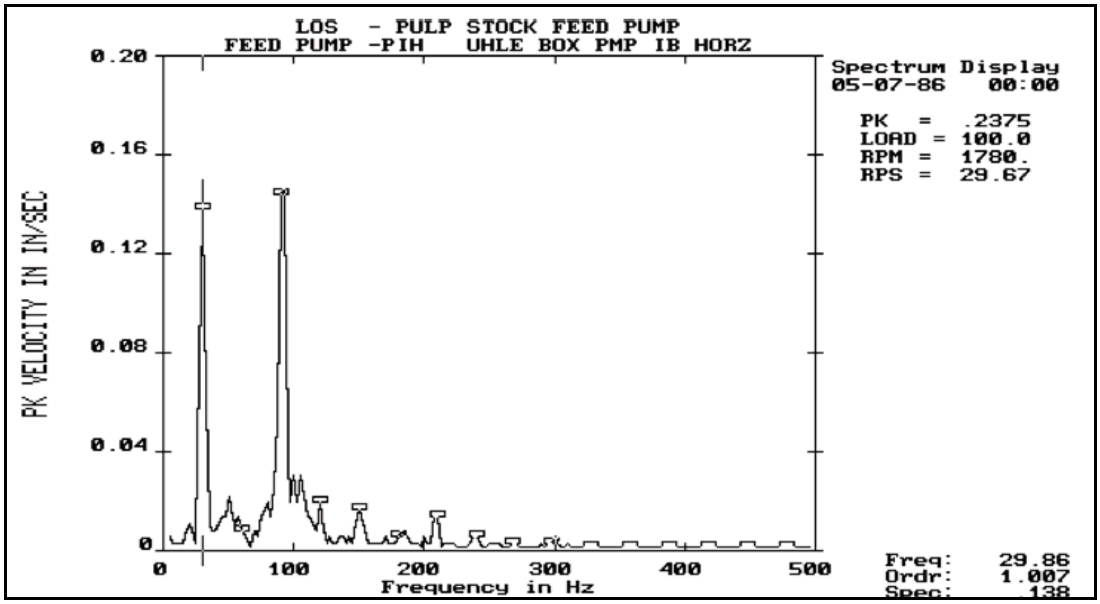


Figure 13

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

By June the overall vibration has diminished. The reduction of 1x and harmonic amplitudes makes the machine appear to be healing itself. If amplitude were the only parameter being considered, a serious fault could go undetected.

Notice how the energy is being spread out among the harmonics of turning speed. Broad based energy is a strong indicator of equipment damage. (See Figure 14)

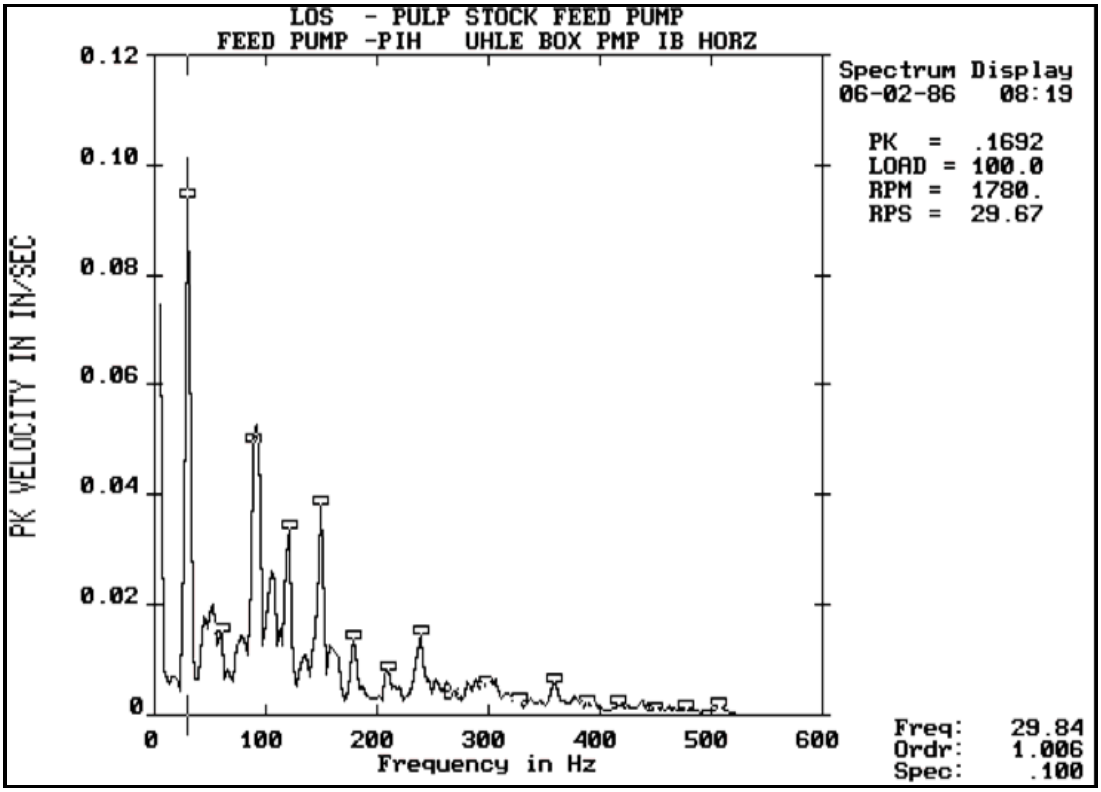


Figure 14

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Turning speed amplitude has not changed much by July. Overall energy has nearly doubled. More harmonics have appeared. The bases of all peaks have broadened. This broadband energy indicates a serious problem. (See Figure 15)

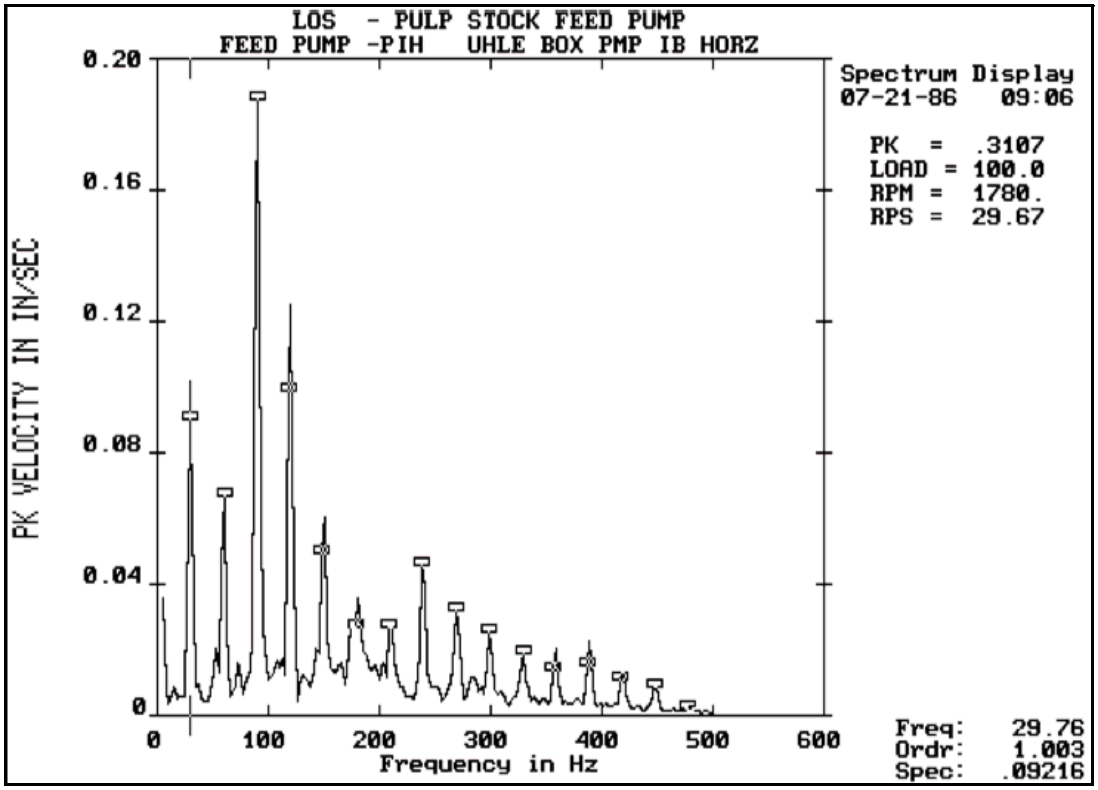


Figure 15

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The scale has more than doubled. Overall energy has nearly doubled from last month. Initially, the broadband energy appears to have disappeared. However, the scale is so high that those amplitudes are somewhat suppressed, giving the appearance that the problem is not as serious as previously thought. (See Figure 16
)

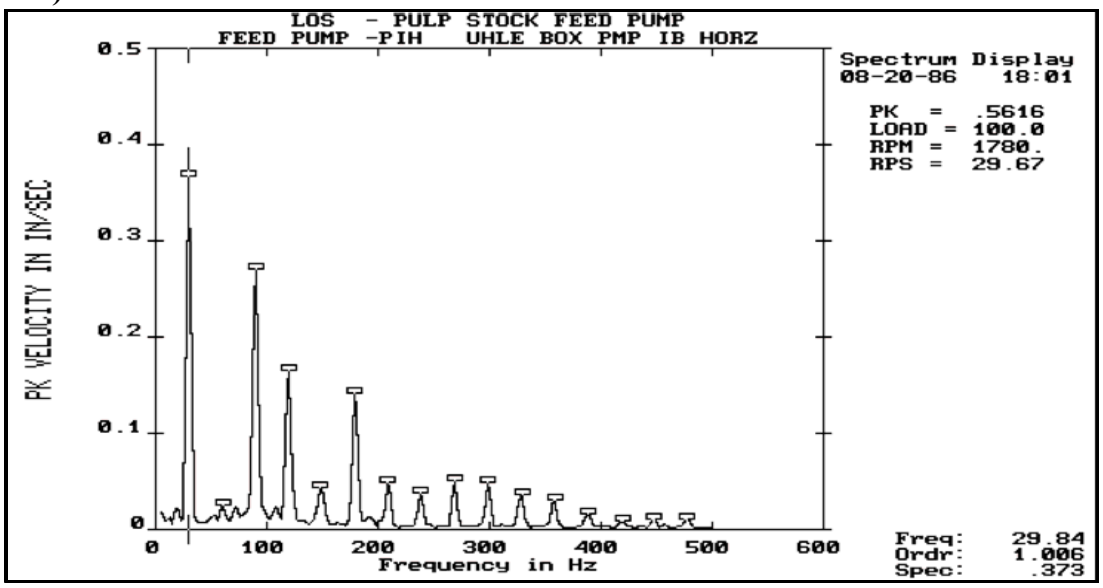


Figure 16

Two months later the scale is reduced by a factor of more than three. Overalls are nearly half of what they were. Harmonics up to 15x are present with a raised noise floor. (See Figure 17)

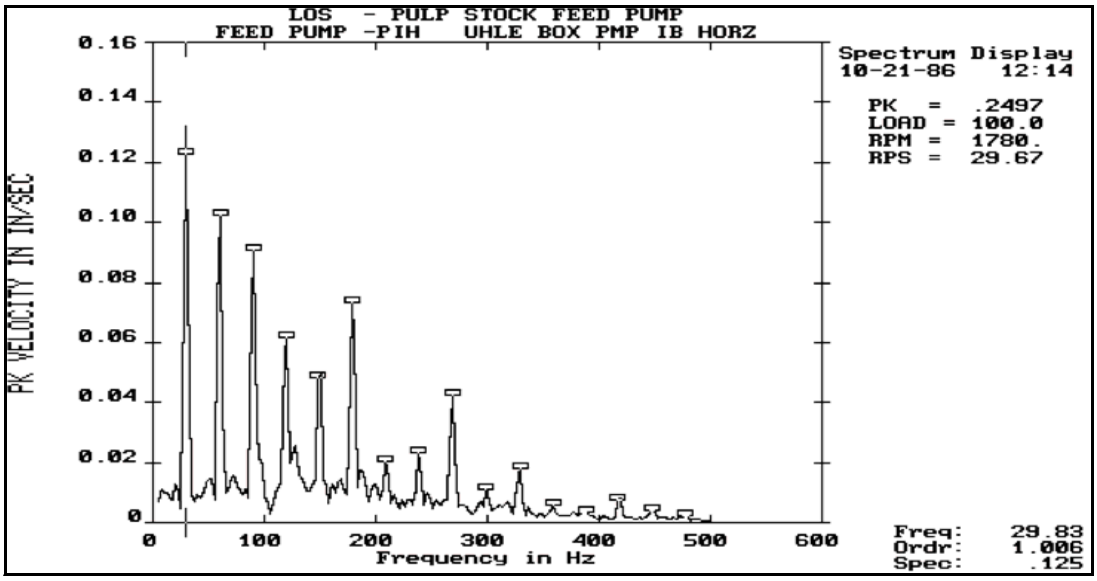


Figure 17

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

1x and overalls have increased by a factor of five (5) by November and fractional harmonics are present. The presence of fractional harmonics indicate a problem regardless of amplitude. (See Figure 18)

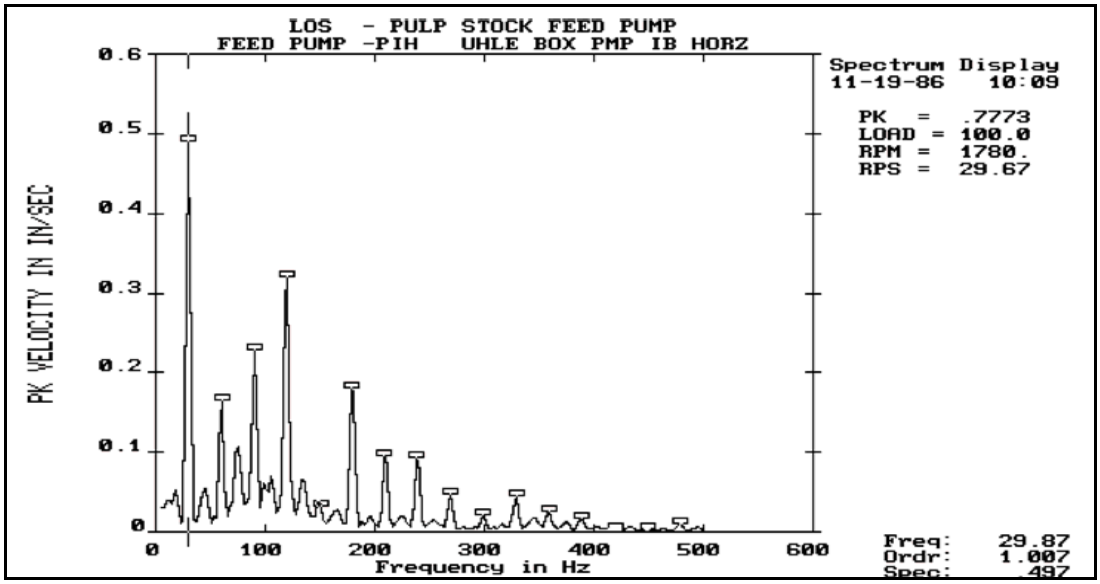


Figure 18

Turning speed is the dominant peak, with harmonics out to the Fmax. Overalls exceed 1”/sec. Catastrophic failure is imminent.

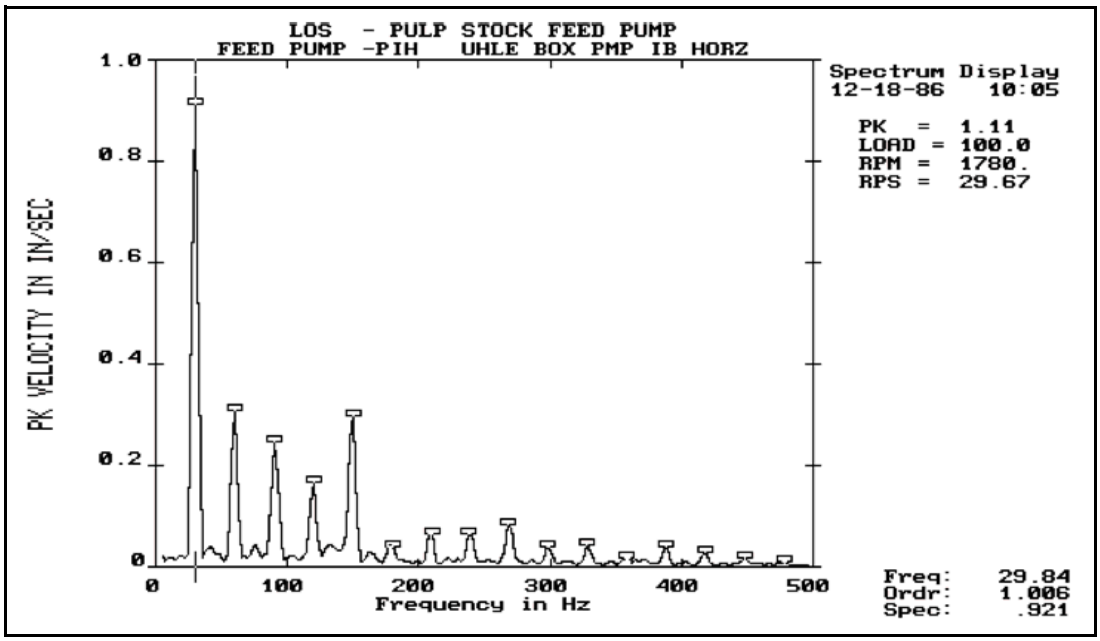


Figure 19

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The trend of overall energy shows an erratic pattern. This usually indicates severe changes in structure integrity, whether in the mounting fixtures or the container of the rotating element in question. What began as a simple unbalance problem has advanced to a major rebuild and costs are multiplied considering the man-hours, material, and loss of production due to downtime. (See Figure 20)

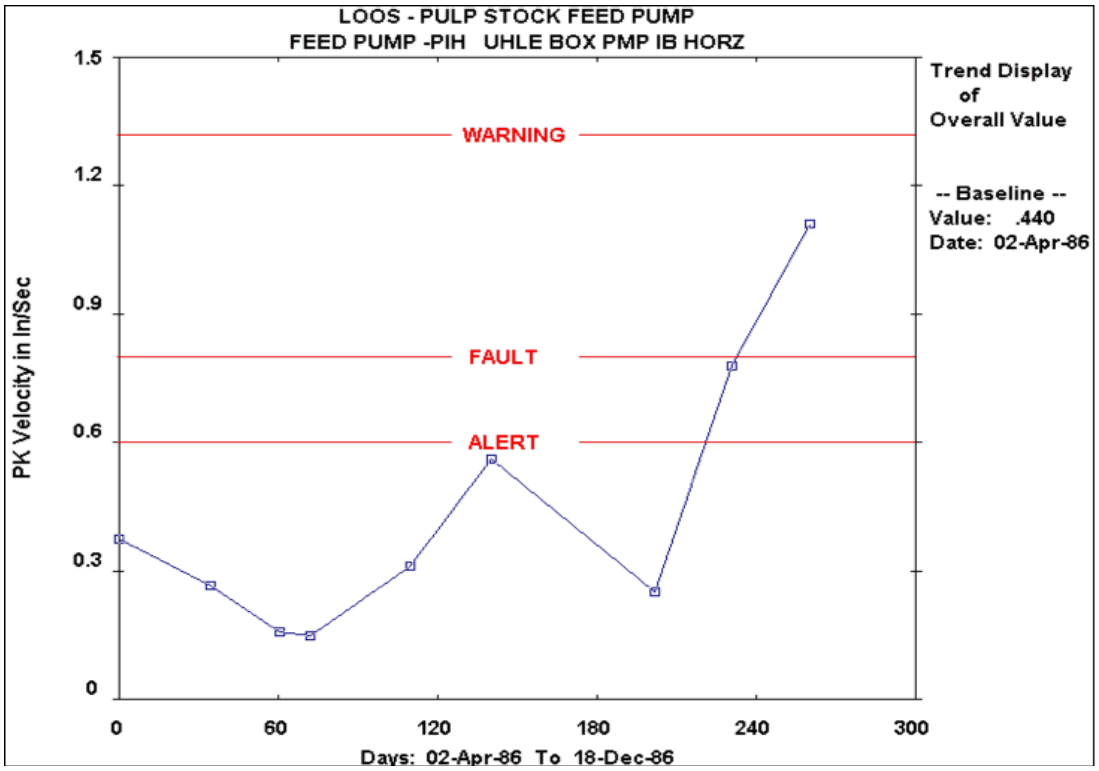


Figure 20

Data Analysis

- Dominant turning speed with harmonics
- Broad band energy with fluctuating overalls
- Strongest in the radial directions.

Diagnosis:

Structural looseness, either in the base or the rotating element housing

Corrective Action:

Balance the equipment, inspect base for loose fasteners, and inspect housings for damage.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #4 - Vertical Pumps

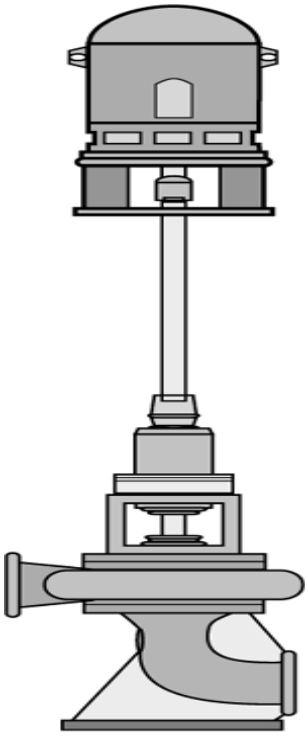


Figure 21

Observations

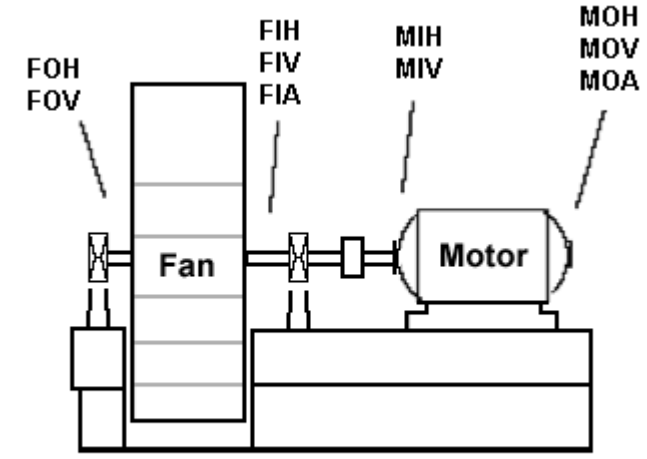
Defects generally appear at 1xTS in the radial direction. The nature of the defect (misalignment, etc.) is a secondary consideration. You can often treat as unbalance any problem that causes a 1xTS peak and extend the life of the pump. Serious diagnostics could well involve disassembly and repair that would not be cost effective.

Looseness

This defect often appears at exactly 1/2x TS in the axial direction. If the amplitude of the peak at 1/2x TS exceeds one-half the amplitude of the 1xTS peak, then you almost certainly have a looseness problem. Note also that axial amplitudes should not exceed one-half the level of radial amplitudes.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #5 - Phase 1 Stack Fan



Direct Drive Centerhung Centrifugal Fan

Figure 22

◆ 250 Horsepower Motor
◆ Center Hung Fan
◆ Pillowblock Bearings on Fan

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Looseness can be directional. When mounting bolts become loose on a horizontal (vertical mounting bolts) mounted machine, the looseness will appear more in the vertical direction. The data in Figure 23 is from the Fan Inboard Vertical position, a typical spectrum of a machine with structural looseness.

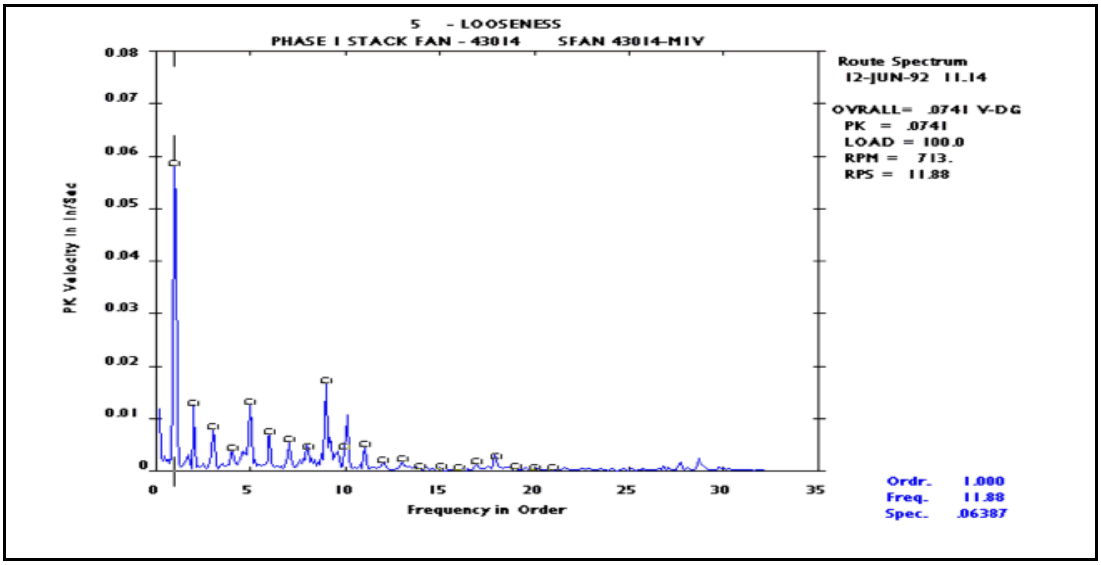


Figure 23

This Time Waveform is also representative of a Looseness pattern. There is a non-repetitive type pattern. Note the relatively low G swing in the Waveform. This is also very typical of a Looseness problem.

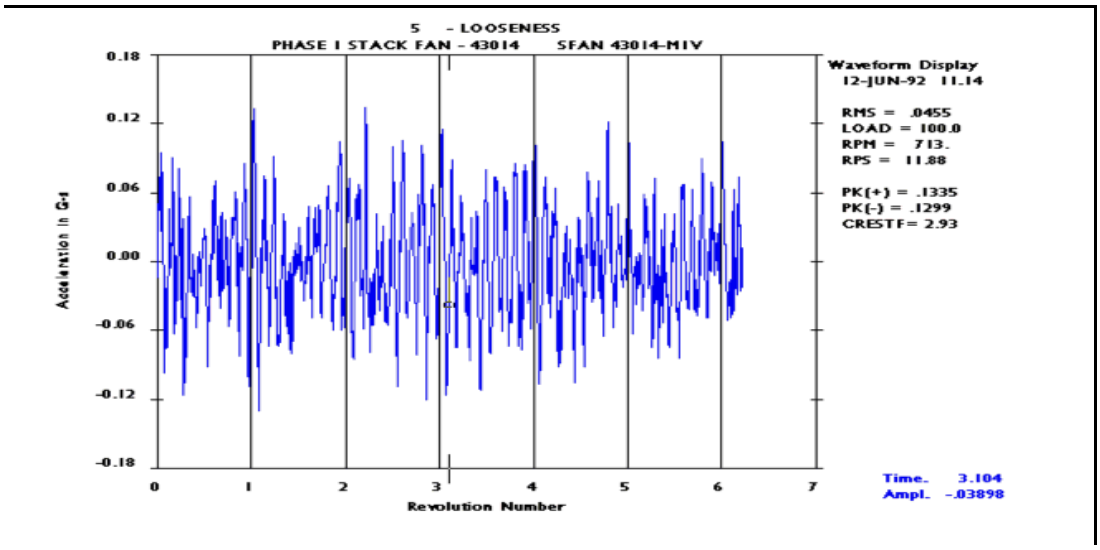


Figure 24

Phase data on this machine would reveal very erratic or unstable data.

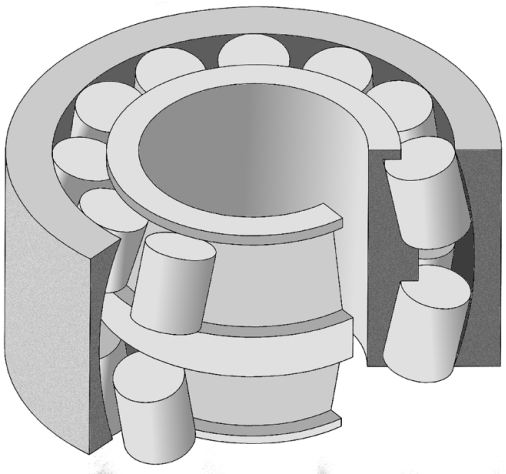
Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Rolling Element Bearings

Section 5

Objectives

- Define a bearing defect
- Determine some causes of premature bearing failure
- Identify the characteristics and signatures of rolling element bearings.
- Determine some corrective actions for bearing problems
- Evaluate the failure progression and severity of rolling element bearings.



Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Rolling Element Bearings

Definition:

A rolling element (also called anti-friction) bearing defect is sometimes difficult to define. A defect can be classified as an imperfection in one or more of the contact surfaces in the bearing. These defects can be virtually invisible to the naked eye, frequently even under a microscope. The problem may be more complex than just an imperfection itself. Often the wrong bearing or lubricant is used, or the bearing may be improperly loaded.

Root Causes of Bearing Failures

Some common causes of bearing failure:

- 43% - Improper lubrication (over and under)
- 27% - Improper mounting methods (hammer, welding, etc.)
- 21% - Other sources (e.g. improper application, manufacturing defects, excessive vibration before and after installation)
- 9% - Normal life expectancy
- Statistics indicate that imbalance and misalignment account for up to 90% of premature bearing failure
- Several sources report that about 10% of all bearings are defective before installation.

Historically, bearings were thought to fail due to lack of lubrication. The reason was that if the bearing was not changed before it failed catastrophically, all the lubrication media escaped. Therefore, lack of lubrication was to blame.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Pattern Recognition

Waveform:

- Sharp impacting “spikes”
- Random impacting with a looseness condition in the bearing
- Excessive “g swing”

Spectrum:

- High frequency / low amplitude peaks
- Harmonics of non-synchronous peaks
- Broad band energy “humps”
- Turning speed sidebands appearing around the rotating face frequency

Some characteristics and general information:

Depending on the type and location of the bearing, one direction may be more useful for analysis than either of the other two. For standard radial loaded ball or roller bearings, the radial locations are usually going to be the best locations for data collection. Tapered cup and cone and thrust bearings generate more axial energy. Therefore the axial direction is going to be best.

- BPFI (Ball Pass Frequency Inner) often falls between 4xTS and 16xTS
- BPFO (Ball Pass Frequency Outer) often falls between 2xTS and 10xTS
- View the spectrum on acceleration to see the high frequency peaks

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Bearing Fault Modes

Vibration analysis can detect the following fault modes on rolling element bearings:

- Defects on raceways
- Defects on rolling elements
- Defects on cage
- Looseness in housing
- Excessive internal clearance
- Bearing turning on shaft
- Misaligned or cocked bearing
- Lack of lubrication

$$Ball\ Pass\ Frequency\ Outer = BPFO = \#of\ rollers \times shaft\ TS \times 0.4x$$

$$Ball\ Pass\ Frequency\ Inner = BPFI = \#of\ rollers \times shaftTS \times 0.6x$$

$$\frac{BPFI}{BPFO} = 1.5(1.4\ to\ 1.6)$$

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Fundamental Defect Frequencies

Example 1

The **inner race is rotating** and the **outer race is stationary**. This is the most common industrial application.

$$FTF = \frac{F}{2} \times \left(1 - \frac{Bd}{Pd} \times \cos\theta\right)$$

$$BPFI = \frac{Nb}{2} \times S \times \left(1 + \frac{Bd}{Pd} \times \cos\theta\right)$$

$$BPFO = \frac{Nb}{2} \times S \times \left(1 - \frac{Bd}{Pd} \times \cos\theta\right)$$

$$BSF = \frac{Pd}{2Bd} \times S \times \left[1 - \left(\frac{Bd}{Pd}\right)^2 \times (\cos\theta)^2\right]$$

Where:

RPM	**=**	revolutions per minute
S	**=**	speed - revolutions per second
FTF	**=**	fundamental train (cage) frequency
BPFI	**=**	ball pass frequency of the inner race
BPFO	**=**	ball pass frequency of the outer race
BSF	**=**	ball spin frequency
Bd	**=**	ball or roller diameter
Nb	**=**	number of balls or rollers
Pd	**=**	pitch diameter
cos	**=**	cosine
θ	**=**	contact angle

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Example 2

The **inner race is stationary** and **outer race is rotating** (e.g., front wheels of some cars).

$$FTF = \frac{S}{2} \times \left(1 + \frac{Bd}{Pd} \times \cos\theta\right)$$

$$BPFI = \frac{Nb}{2} \times S \times \left(1 - \frac{Bd}{Pd} \times \cos\theta\right)$$

$$BPFO = \frac{Nb}{2} \times S \times \left(1 + \frac{Bd}{Pd} \times \cos\theta\right)$$

$$BSF = \frac{Pd}{2Bd} \times S \times \left[1 - \left(\frac{Bd}{Pd}\right)^2 \times (\cos\theta)^2\right]$$

Where

RPM	**=**	revolutions per minute
S	**=**	speed - revolutions per second
FTF	**=**	fundamental train (cage) frequency
BPFI	**=**	ball pass frequency of the inner race
BPFO	**=**	ball pass frequency of the outer race
BSF	**=**	ball spin frequency
Bd	**=**	ball or roller diameter
Nb	**=**	number of balls or rollers
Pd	**=**	pitch diameter
cos	**=**	cosine
θ	**=**	contact angle

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Bearing Load Life Formulas

$$H = \left(\frac{C}{L}\right)^3 \times \left(\frac{16667}{RPM}\right)$$

Where:

H	**=**	bearing life in hours
C	**=**	manufacturer's bearing capacity in lbs
L	**=**	actual bearing load in lbs.
RPM	**=**	shaft speed in revolutions per minute

You can further decrease bearing life by increasing:

- Load (cubed effect)
- Speed

Load has the most adverse effect on bearing life.

How Does Vibration Affect Bearing Life?

$$H = \left(\frac{C}{L + 6.7753 \times 10^{-5} MVF}\right)^3 \times \left(\frac{16667}{RPM}\right)$$

Where:

H	**=**	ball bearing life in hours
C	**=**	manufacturer's bearing capacity in lbs.
L	**=**	in-service bearing load in lbs.
M	**=**	weight in lbs. of mass opposing vibration
V	**=**	velocity of vibration in IPS
F	**=**	frequency of vibration in CPM or RPM

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Example:

- Dead load = 1000 lbs.
- RPM = 1800
- Bearing capacity = 20,000 lbs.
- Mass = 13,000 lbs.

Vibration in IPS	Bearing Load in lbs.	Bearing Life	Percent of Life (Compared to Life @0.2 IPS)
0	1000	8.46 years	228%
0.2	1316	3.70 years	100%
0.4	1633	1.94 years	52%
0.6	1950	1.15 years	31%
1.0	2584	5.6 months	13%
1.5	3376	2.5 months	6%
2.0	4169	1.4 months	3%
3.0	5754	2.3 weeks	1.1%

Note
In this example, bearing life at a vibration of 1.0 IPS is 13% of that for 0.2 IPS.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Formulas for Approximating Unknown Bearings

Exact bearing data may not always be available. Use the formulas below to help calculate approximate frequencies for bearing faults.

$$FTF = 0.4 \times RPM$$

$$BPFO = 0.4 \times N \times RPM$$

$$BPFI = 0.6 \times N \times RPM$$

$$\frac{BPFI}{BPFO} = 1.5$$

Where:

N	**=**	number of rollers

- For motors, pumps, fans, compressors, etc., estimate 7 to 16 rollers.
- For large rolling mill bearings, estimate more than 16.

Examples

An SKF 22228 bearing has 19 balls and a shaft turning speed of 29.6 Hz.

$$Estimated\ BPFI = 19 \times 29.6 \times 0.6 = 337.44 Hz$$

$$Actual\ BPFI = 319.68 Hz (approximately\ 5\%)$$

$$Estimated\ BPFO = 19 \times 29.6 \times 0.4 = 224.96 Hz$$

$$Actual\ BPFO = 243.31 Hz (approximately\ 5\%)$$

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

How Long Will the Bearing Last?

What is the history and the current condition of the bearing?

- Size and number of defects
- Defective components (roller/cage)
- Loss of internal geometry
- Rate of progression

Why is the bearing failing?

- Loss of internal clearance
- Loss of lubricant
- Excessive external vibration

How long has it been in service?

- Proportional to how long it has been running and when defects first appeared

What is the speed of the unit?

- 3600 RPM and above can fail quickly
- 300 RPM and below can go for several months

What is the previous experience with similar equipment and similar failures?

- Beware - no two identical cases exist

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Evaluating Failure Progression and Severity

- Bearing components normally fail in the following order: race defects, ball or roller defects, cage defects (unless the bearing was defective when installed).
- Inner race defects and failures occur at much lower amplitudes than outer race defects.
- Early faults generate predicted defect frequencies and harmonics, frequently for only one of the races.
- Extended harmonics of the defect frequency may indicate multiple defect sites or extended defect size.
- Appearance of defect frequencies generated by other components indicates progressive damage. The cage is usually the last component to fail and can result in wide shifts in frequency or audible noises just before seizure.
- Race defect frequencies are modulated by the shaft speed, which results in the appearance of sideband peaks. The number of sideband peaks increases as the damage progresses.
- Loss of individual peaks and/or significant broadband energy indicates significant changes in the bearing geometry.
- Inadequate lubrication can result in very accelerated failure rates and should be corrected immediately.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Analysis Parameters and Alarm Limits

Table 1 gives general guidelines for analysis parameter sets for measurement points on rolling element bearings. Define a baseband frequency range of 65xTS with a minimum of 800 lines of resolution.

Band Description	Frequency Range
Subsynchronous and 1xTS	0.0 and 1.5xTS
2xTS	1.5xTS to 2xTS
3xTS to 8xTS	2.5xTS to 8xTS
1st bearing band	8.5xTS to 35.5xTS
2nd bearing band	35.5xTS to 65xTS
High frequency band	1kHz to 20 kHz

Table 1

Alarm Limits

Table 2 gives general guidelines for alarm limit sets for measurement points on rolling element bearings.

	Alert	Fault
Overall	0.3 IPS	0.5 IPS
Sub and 1x	0.25 IPS	0.4 IPS
2x	0.15 IPS	0.3 IPS
3 to 8x	0.12 IPS	0.2 IPS
1st bearing band	0.04 IPS	0.06 IPS
2ns bearing band	0.05 IPS	0.08 IPS
High frequency	3.0 g's	7.0 g's

Table 2

These general alarm amplitude values are usually acceptable for machines running above 1000 RPM. Use experience with bearing degradation on specific machine types to adjust the recommended alarm levels accordingly.

On slow speed machine, bearings that failed have shown peak amplitudes as low as 0.01 to 0.04 IPS.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Typical Patterns of Normalized Bearing Frequencies

Bearing Series	Number of Elements	FTF	BSF	BPFO	BPFI
6403	6	0.335	1.356	2.101	3.960
6405	7	0.361	1.656	2.526	4.500
6407	7	0.361	1.656	2.526	4.500
6409	7	9.361	1.656	2.526	4.500
22324S	14	0.402	2.382	5.628	8.340
22328S	14	0.402	2.382	5.628	8.340
22332	14	0.401	2.358	5.616	8.340
22336	14	0.403	2.418	5.646	8.340
22340	14	0.403	2.412	5.640	8.340
22348	16	0.415	2.772	6.630	9.360
22356	17	0.415	2.844	7.080	9.900
23022S	26	0.448	4.644	11.634	14.34
23023S	26	0.448	4.698	11.646	14.34
23030S	26	0.451	5.040	12.186	14.82
23034S	28	0.451	4.986	12.624	15.36
23038S	28	0.451	4.962	12.618	15.36
23044	27	0.449	4.788	12.120	14.88
23052	28	0.448	4.680	12.090	14.88
23060	28	0.448	4.728	12.102	14.88
23068	28	0.449	4.860	12.138	14.88
23076S	30	0.454	5.316	13.614	16.38

Table 3

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Antifriction Bearing

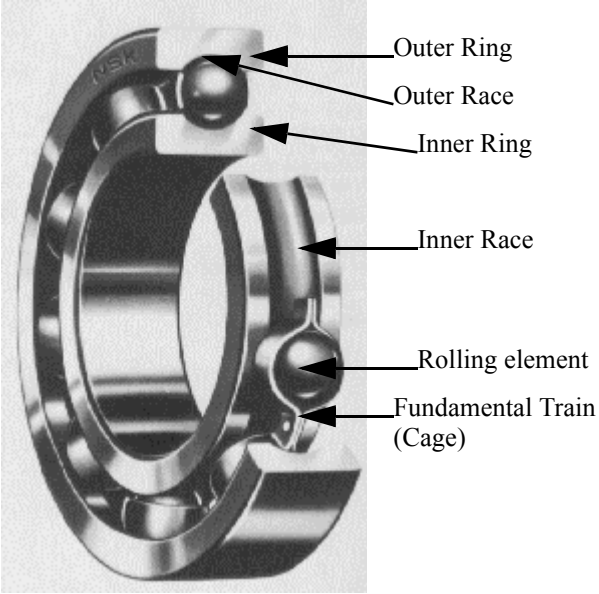


Figure 1

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #1 - Tenter Zone Exhaust Fan

A significant amount of sub-synchronous energy is present in the data from May to November. The higher frequency energy increased considerably starting in September. Inspect individual spectra to try to determine the source of the vibration and the severity of the problem. (See Figure 2)

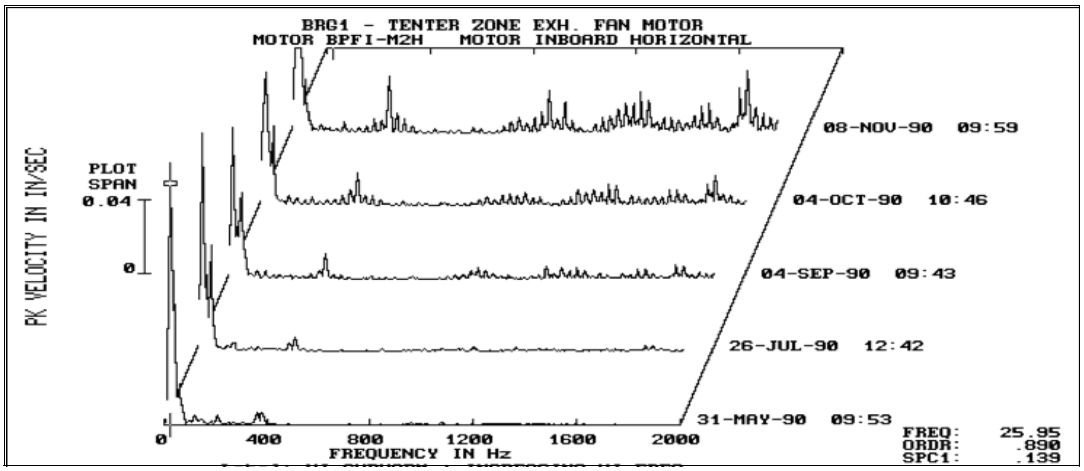


Figure 2

The spectrum in Figure 3 was taken from a belt drive unit. The dominant peak was determined to be a function of the belt, actually 2X belt frequency. Even viewing the data in velocity, the high frequency, non-synchronous energy is significant.

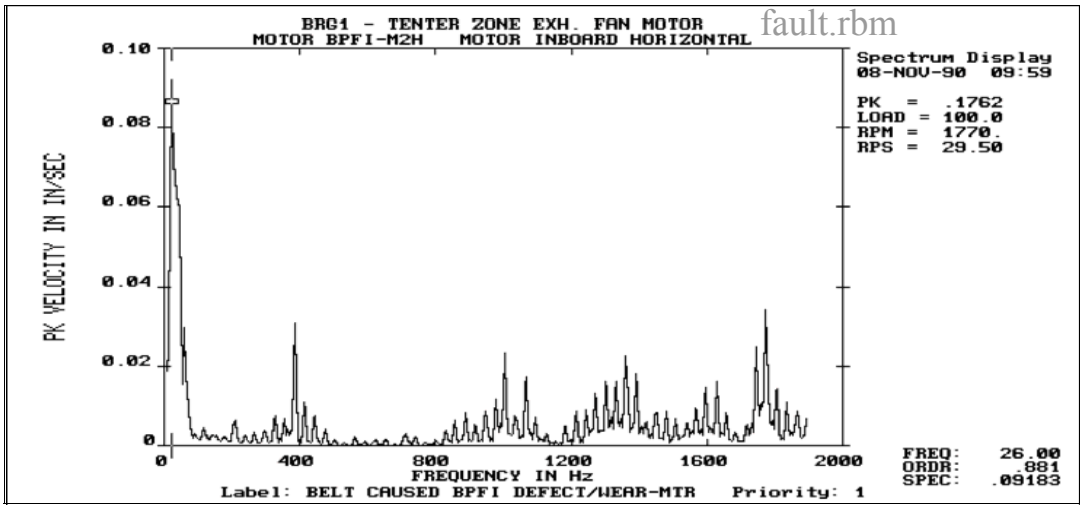


Figure 3

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

LIST OF SPECTRAL PEAKS
****

Machine: (BRG1) TENTER ZONE EXH. FAN MOTOR
Meas. Point: MOTOR BPFI-M1H --> MOTOR OUTBOARD HORIZONTAL
Date/Time: 08-Nov-90 09:57:55 RPM= 1763. Units=In/Sec PK
Data Label: 133.2 HZ IS INBOARD BRG BPFI Priority: 1

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	30.97	.0827	1.054	13	1447.07	.0139	49.248
2	43.28	.0503	1.473	14	1476.93	.0273	50.264
3	59.09	.0317	2.011	15	1506.41	.0177	51.267
4	120.02	.0125	4.085	16	1536.89	.0209	52.305
5	383.55	.0142	13.053	17	1566.37	.0105	53.308
6	886.36	.0133	30.165	18	1594.87	.0254	54.278
7	1004.15	.0100	34.174	19	1624.44	.0196	55.284
8	1181.78	.0176	40.219	20	1654.48	.0276	56.307
9	1211.56	.0136	41.233	21	1684.17	.0152	57.317
10	1299.72	.0119	44.233	22	1743.24	.0111	59.327
11	1329.58	.0243	45.249	23	1773.05	.0107	60.342
12	1359.29	.0119	46.260	24	1802.55	.0197	61.346

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.1297	0.0 / 0%	.0317 / 6%	.1251 / 93%
(ELECTRICAL=	.0125 / 1%	FLOOR= .1049	SPEC CREST= 2.93)

Table 4

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

A rolling element bearing defect will frequently generate a pattern in the waveform that resembles a school of angel fish swimming in succession. (See Figure 4)

Distinctive impacting, with modulation and ringing-down is evident. This pattern indicates that the rolling elements are passing over a race defect similar to the tires on a car encountering a series of pot-holes in the road. The most severe impacting occurs when the roller exits the opposite side of the defect, just as the tires produce the highest impacting as they exit the pot-hole.

Improper belt tension probably aggravated an initially minute bearing defect, causing the bearing to fail prematurely. Had the belts been properly tensioned this bearing problem may have been avoided.

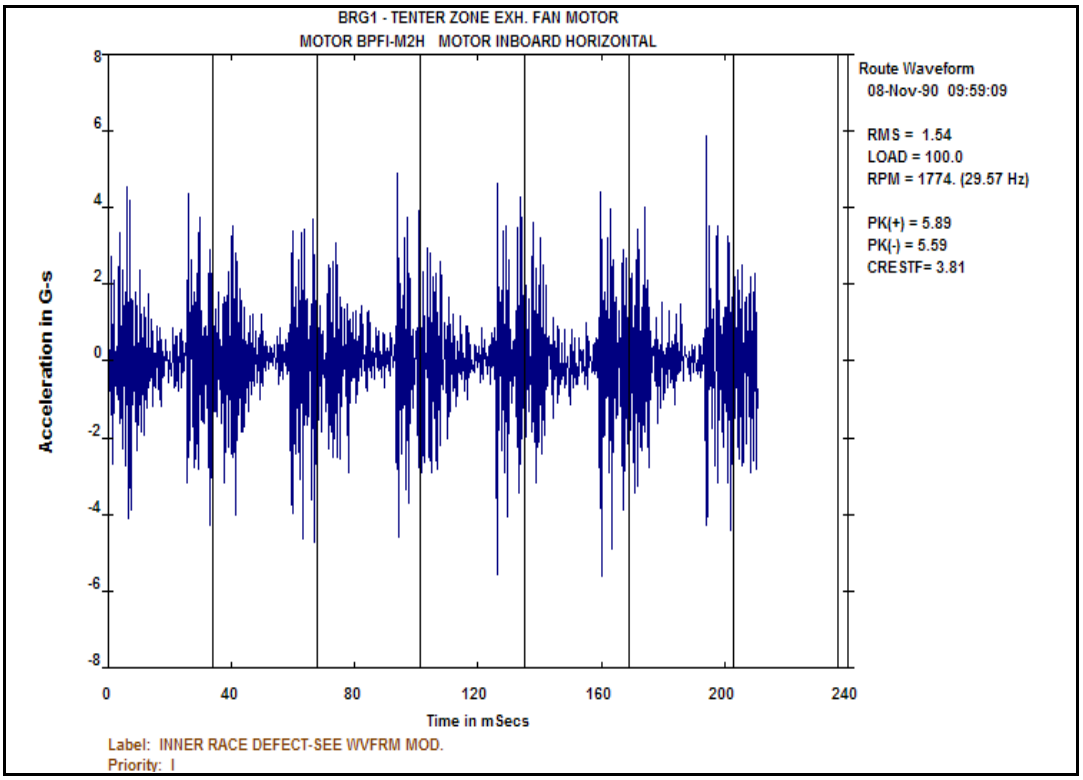


Figure 4

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The spectrum in Figure 5 comes from the Motor Inboard Vertical position. Patterns similar to those found in the horizontal position are present. Mounds of energy, possible turning speed sidebands, with the fault frequency overlays help confirm an inner race defect.

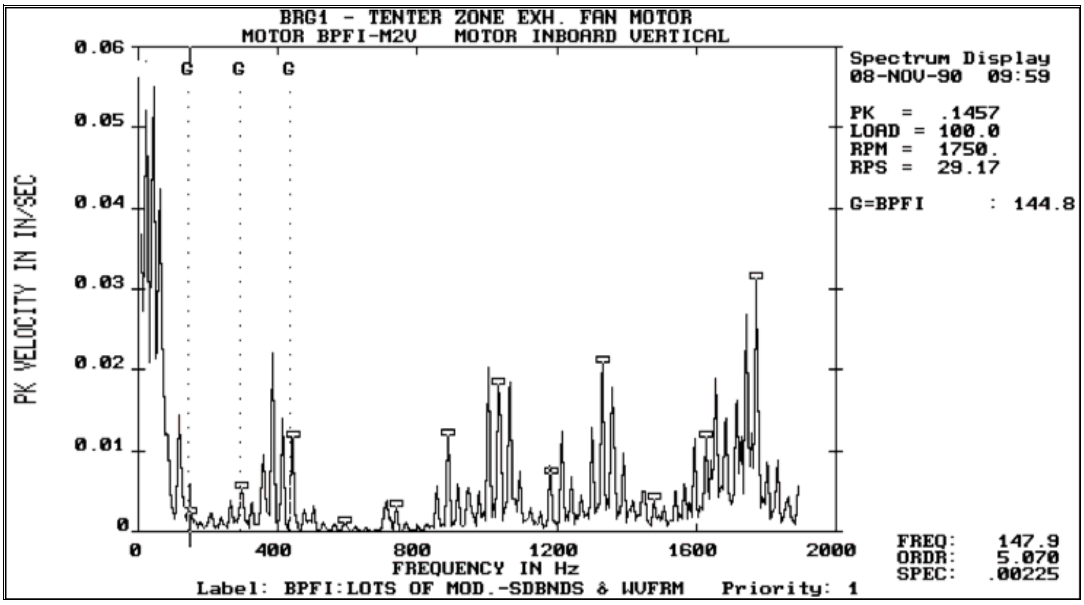


Figure 5

Figure 6 shows the waveform from the vertical position. The high impacting, ringing-down, excessive "g swing", and an "angel fish" pattern all indicate a bearing defect.

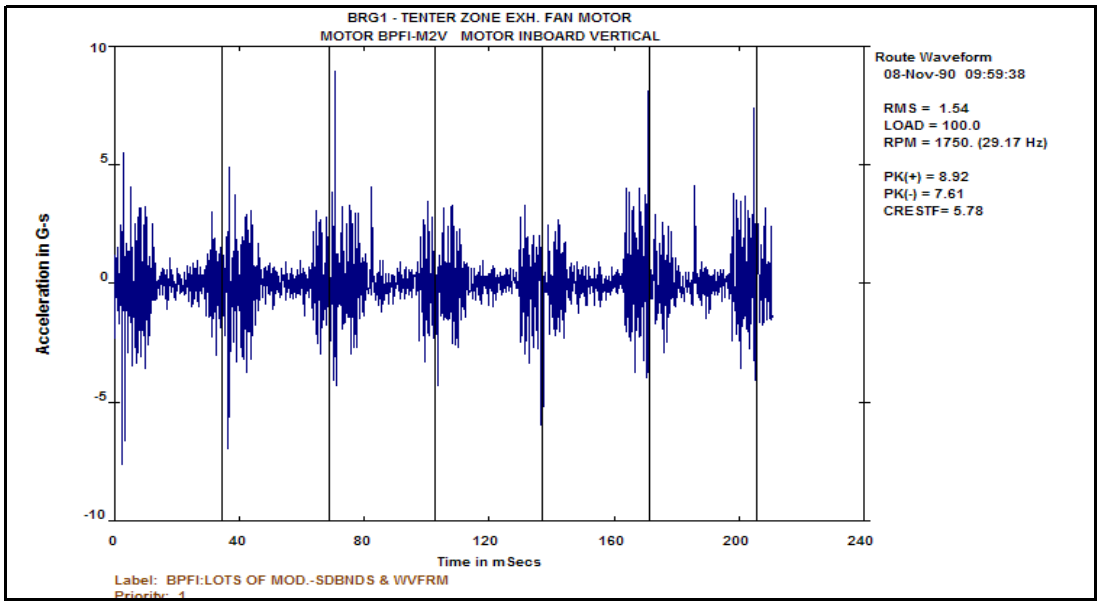


Figure 6

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #2 - Primary Coarse Screen Reject Agitator

The data shown in Figure 7 was taken from a belt-driven coarse screen reject agitator in a recycled paper plant. Non-synchronous energy is dominant, with a major peak at approximately 5 orders in October and November. Blade pass or a bearing defect could possibly be generating the apparent 5xTS energy. That peak is actually at 29.49 Hz, motor shaft turning speed. This condition usually indicates sheave misalignment or run-out. February's data indicates a significant change in the vibration signature. (See Figure 7)

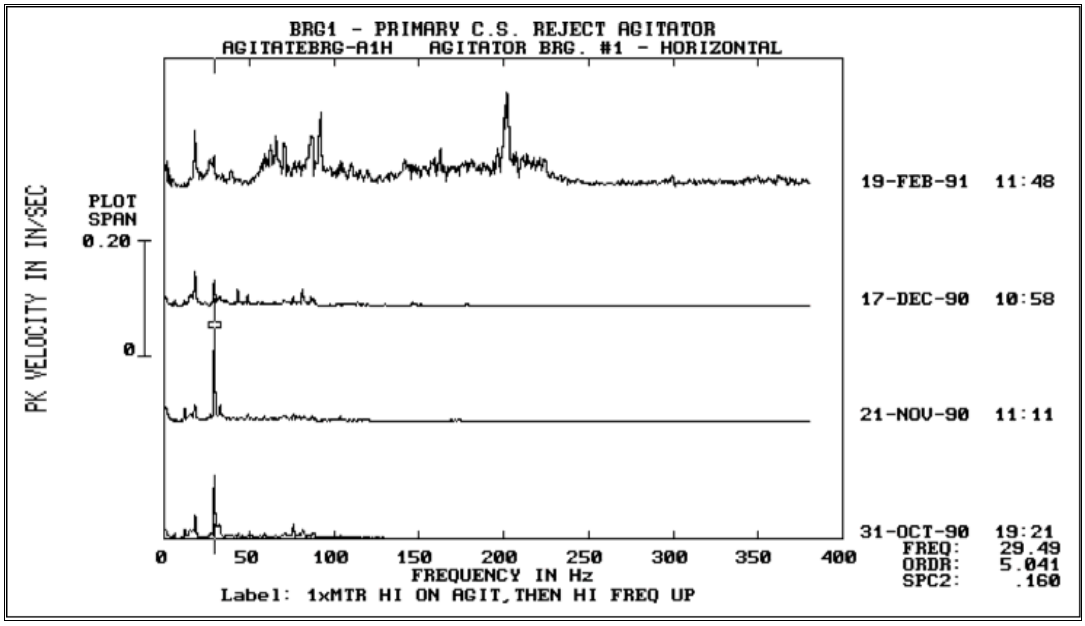


Figure 7

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

By February there is broad band energy almost reaching the Fmax. Even though only 2 peaks exceed 0.1 IPS, the overall vibration amplitude exceeds 0.7 IPS. When there is so much broad band energy in a spectrum, the overall energy levels will almost always be elevated. In this case, a relatively new machine is exhibiting a pattern that indicated a misaligned or eccentric sheave, initially. However, after only a few months the pattern has changed significantly. (See Figure 8)

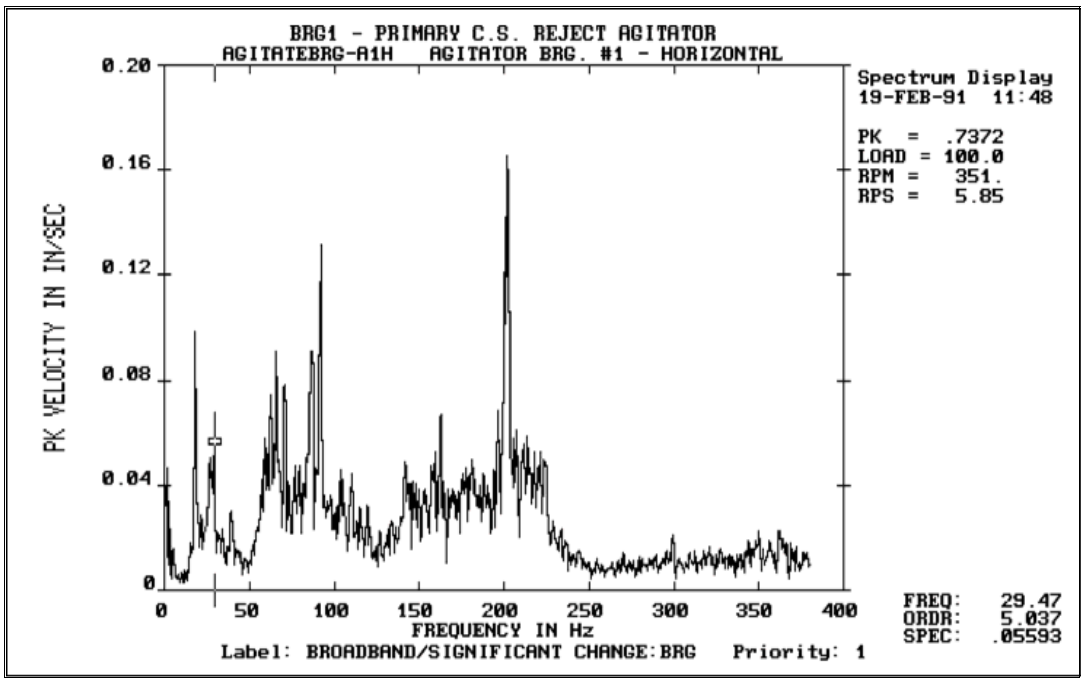


Figure 8

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Significant energy (6 to 10 G swing) is evident in the waveform in Figure 9. With the amount of bearing looseness as is probably in this machine, impacting and modulation are difficult to discern. The Fmax is approximately 400 Hz. Since acceleration accentuates higher frequencies, bearing wear between 50 and 200 Hz is not as apparent as with higher frequencies. Based on the available data, an outer race is suspected.

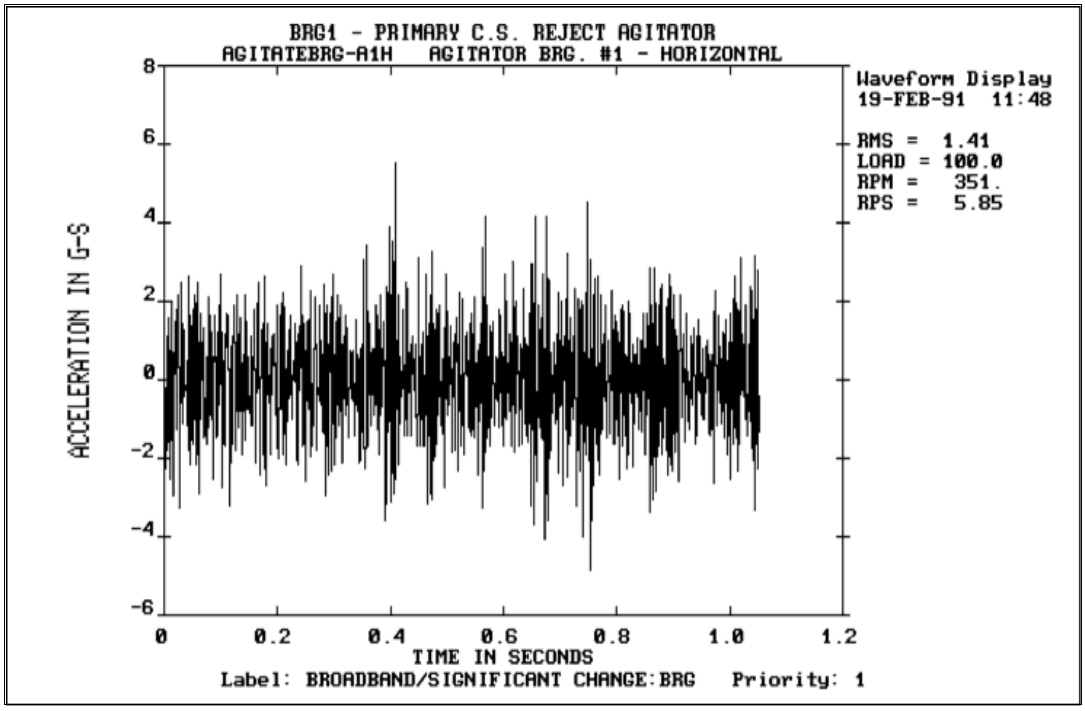


Figure 9

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #3 - Chemical Plant Sludge Pump

Figure 10 is the multi-spectrum plot for the Chemical Sludge Pump. A chronological failure pattern is evident from April to September. April's data does not appear to show any significant problems. This can be misleading because the scale for the multiple spectrum plot is forced to a level high enough to accommodate the highest amplitude(s) of any given peak in the entire display. It is necessary to analyze the individual spectra from each month to properly diagnose the problem(s).

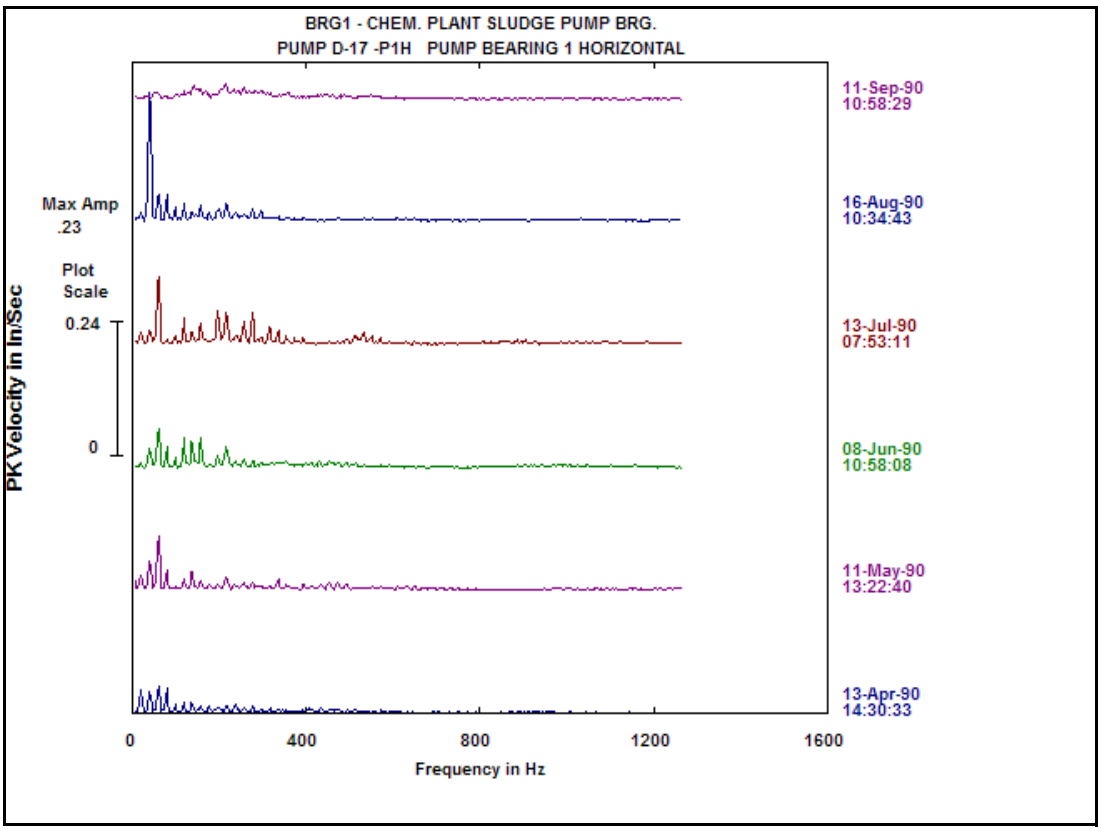


Figure 10

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

April's data (Figure 11) indicates a degree of looseness. Were this structural looseness, higher amplitudes would be expected. With the higher frequencies and the elevated noise floor and relatively low overalls, a bearing defect is suspected.

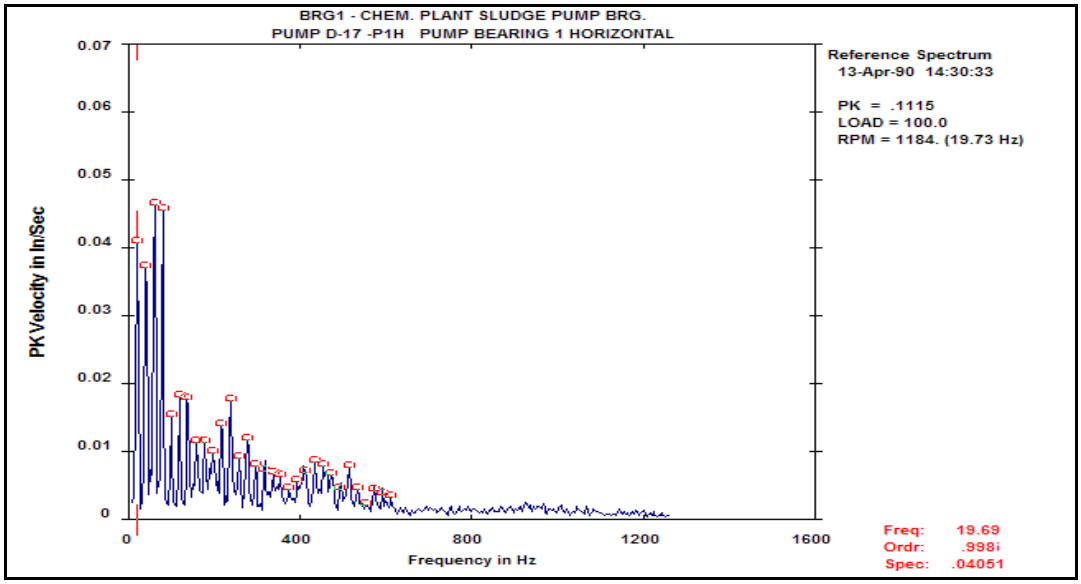


Figure 11

The waveform in Figure 12 is indicating some impacting. A Crest Factor of greater than 3 indicates the impacting is fairly significant. Although there seems to be some repetition of some frequencies, looseness is also suspected because of the random pattern.

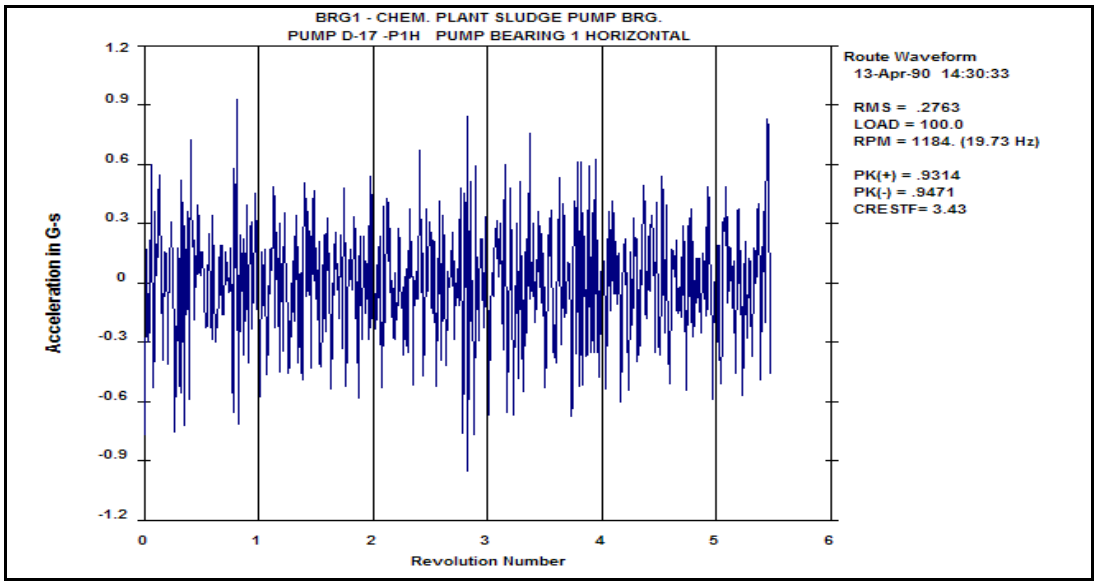


Figure 12

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The overall levels have increased approximately 36%. Turning speed harmonics are more pronounced, further implying a degrading looseness condition. Some higher frequency peaks are starting to appear. This usually means some type of bearing defect is present. (See Figure 13)

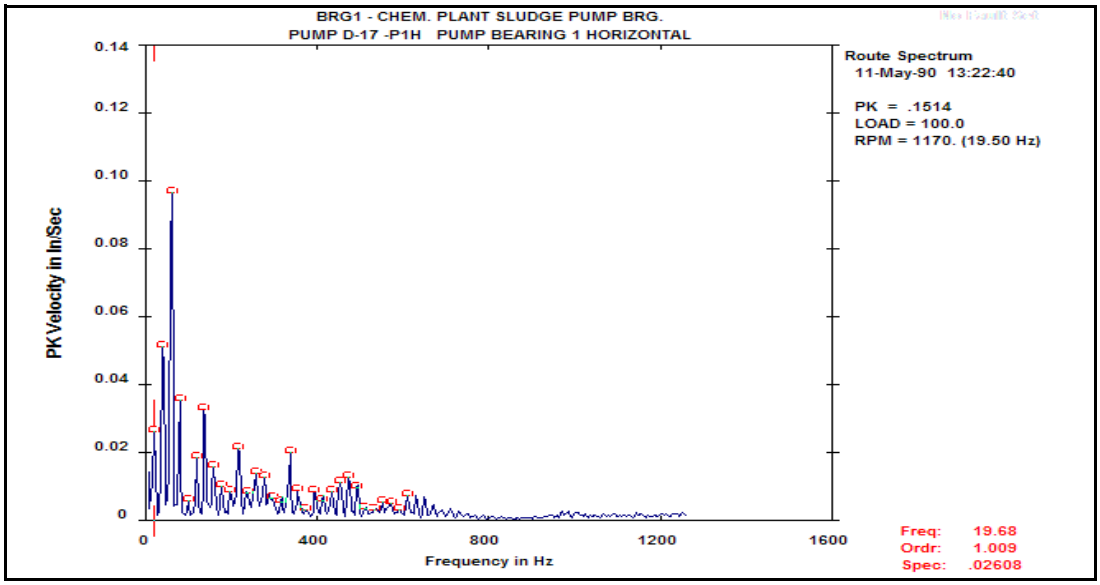


Figure 13

The waveform in May is exhibiting somewhat of an "Angel Fish", ring down pattern, with a 2G swing; further evidence of an impending bearing failure.(See Figure 14)

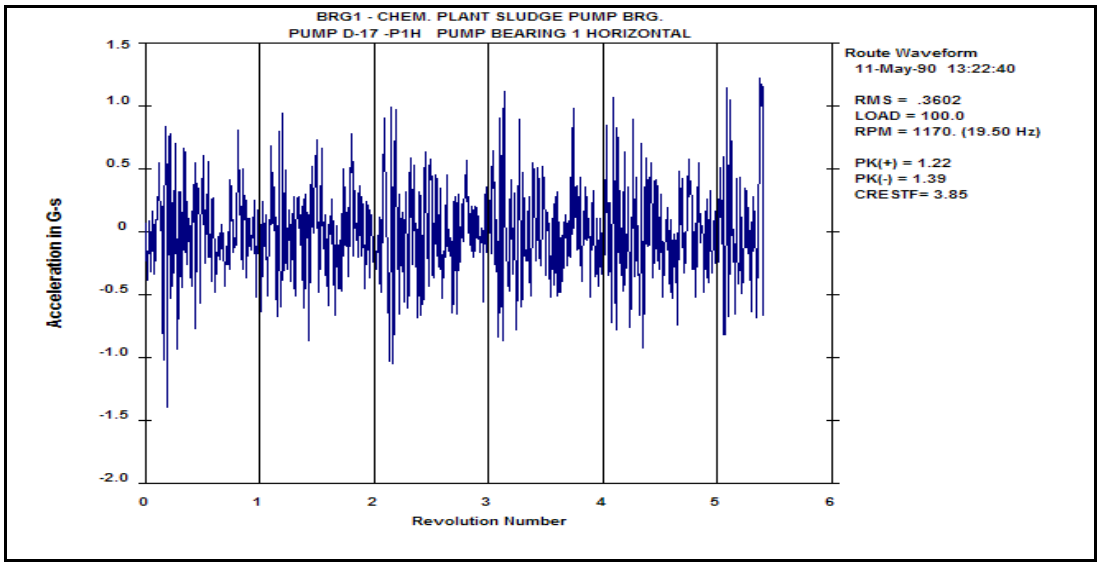


Figure 14

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Increased high frequency, broad band energy in July indicates excessive clearance between the bearing components. The cage and rolling elements are probably wearing out rapidly. (See Figure 15)

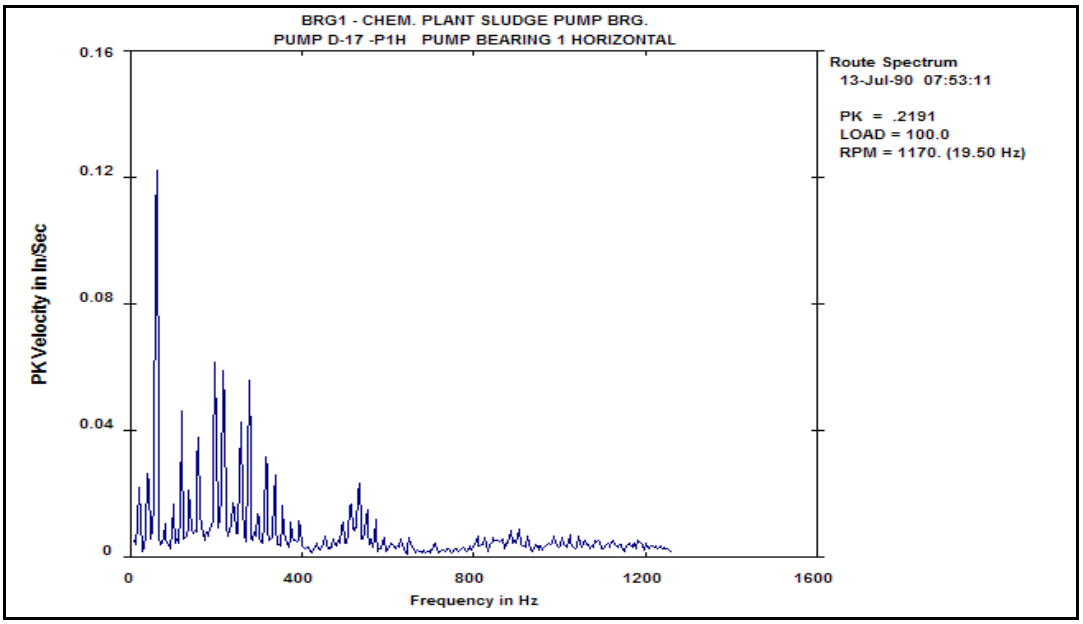


Figure 15

The time domain data in Figure 16 shows high levels of impacting. In just one month, the G swing has increased from 4 to 6 ½. Looseness is indicated by the random pattern.

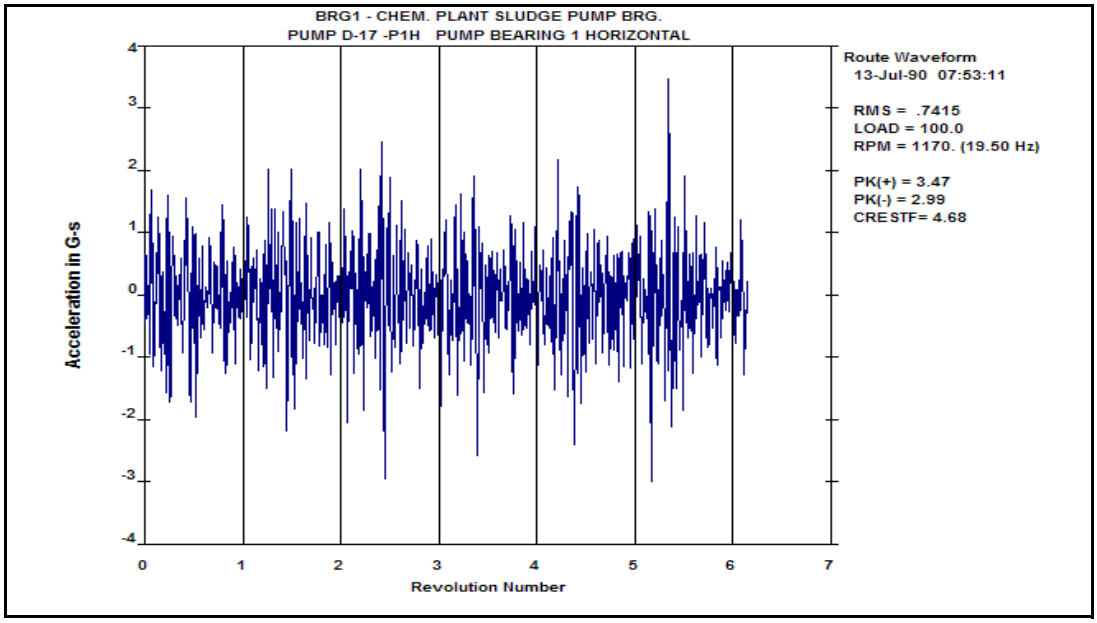


Figure 16

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

A cracked inner race may be responsible for the increased amplitude of the 2xTS peak, raised noise floor, and high frequency energy. The bearing has possibly lost its interference fit on the shaft. (See Figure 17)

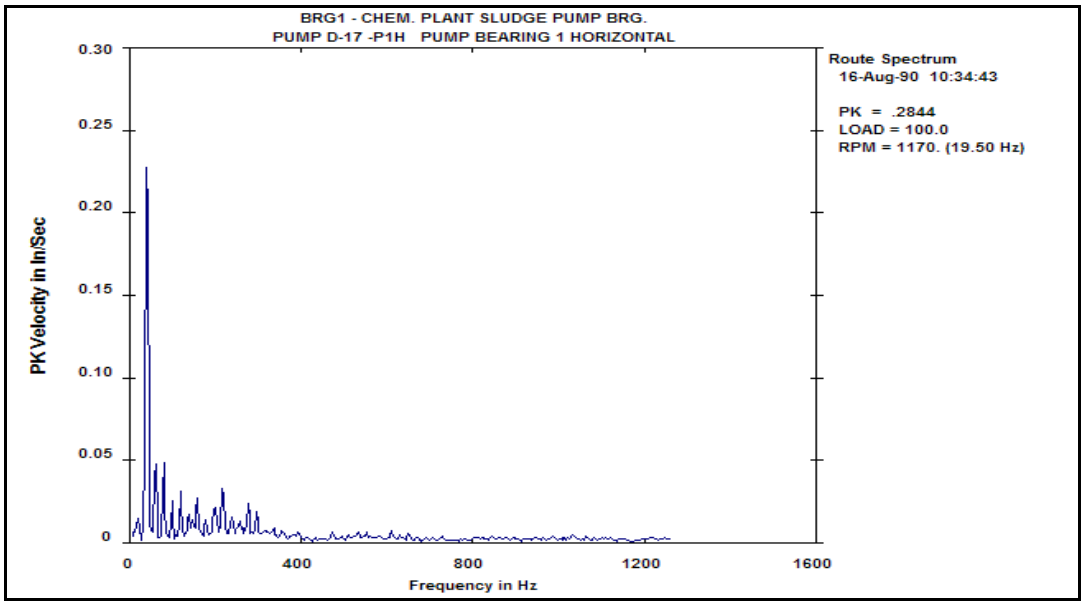


Figure 17

A sudden apparent decrease in the G swing seems to mean the bearing has repaired itself. The bearing has actually begun to fall apart. Now there is more looseness than ever. There is less metal to metal contact, therefore nothing to transmit the energy from the race to the accelerometer. (See Figure 18)

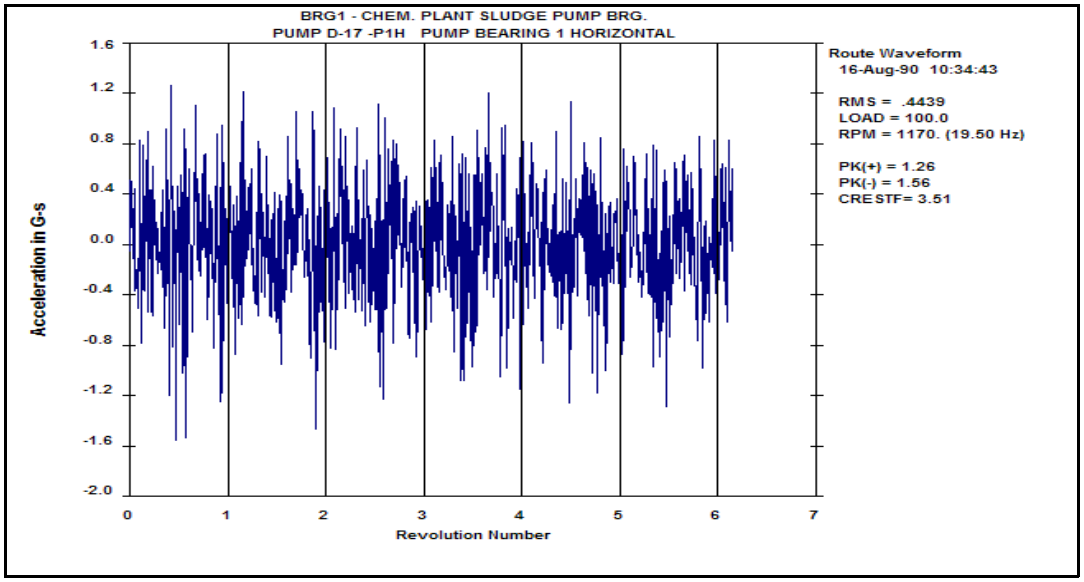


Figure 18

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Considering the relatively low overalls (0.1193 inches per second) and the fact that the highest amplitude peak is below 0.03 inches per second, it may seem that the bearing condition has improved. Actually, the bearing has disintegrated. The presence of extreme amounts of broad band energy in the spectrum and a random waveform with a high G swing and a Crest Factor greater than 3, in this case 5, indicates a significant problem. Relying on discrete peak amplitude alone may lead to missing a severe problem. This bearing fell apart during disassembly. (See Figure 19)

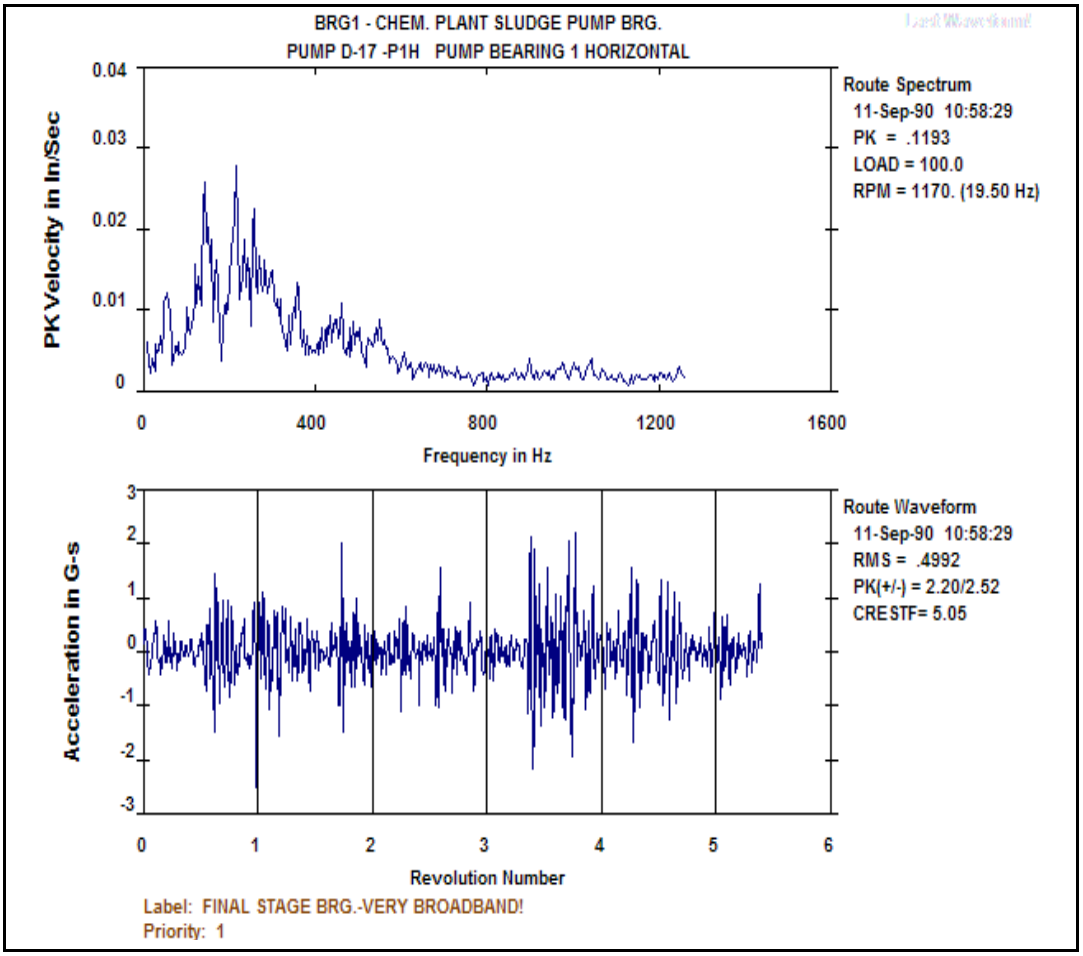


Figure 19

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #4 - Film Trim Takeaway Blower

The multiple spectra plot from this blower shows significant change in the high frequency region in the three months between April and July. The pattern resembles looseness, but the peaks are not harmonics of TS. (See Figure 20)

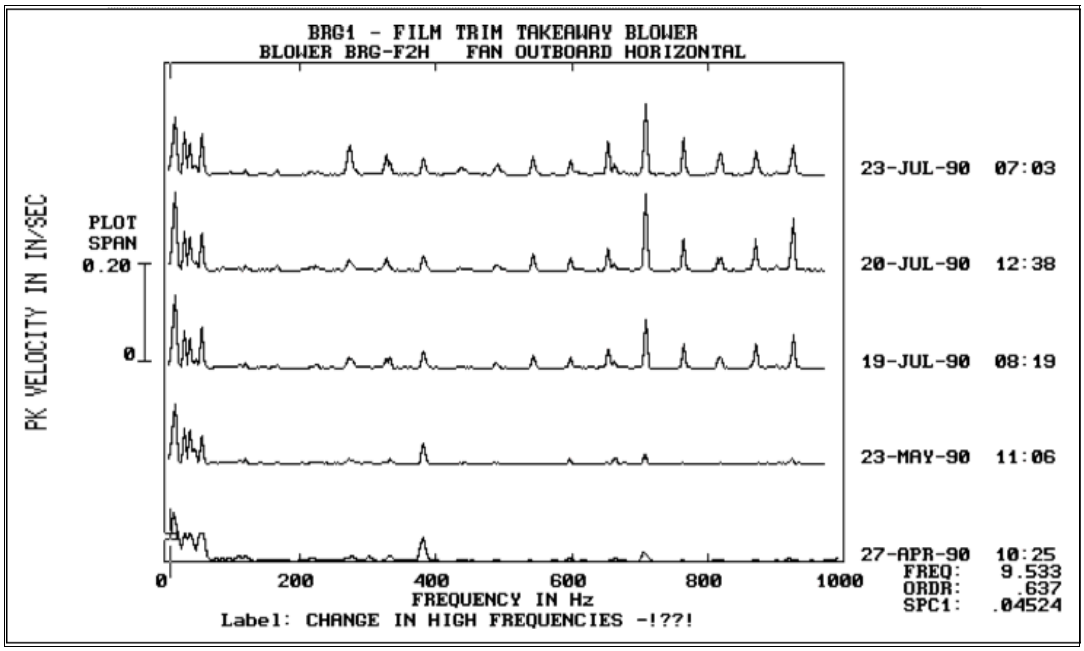


Figure 20

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

This blower had an outer race defect. The spectrum in Figure 21 displays the fault frequency overlays. Use fault frequencies carefully. The harmonic cursors are marking the actual generated bearing frequencies. The geometry of the bearing has changed. Therefore the frequencies also change. Fault frequency lines do not line up very well.

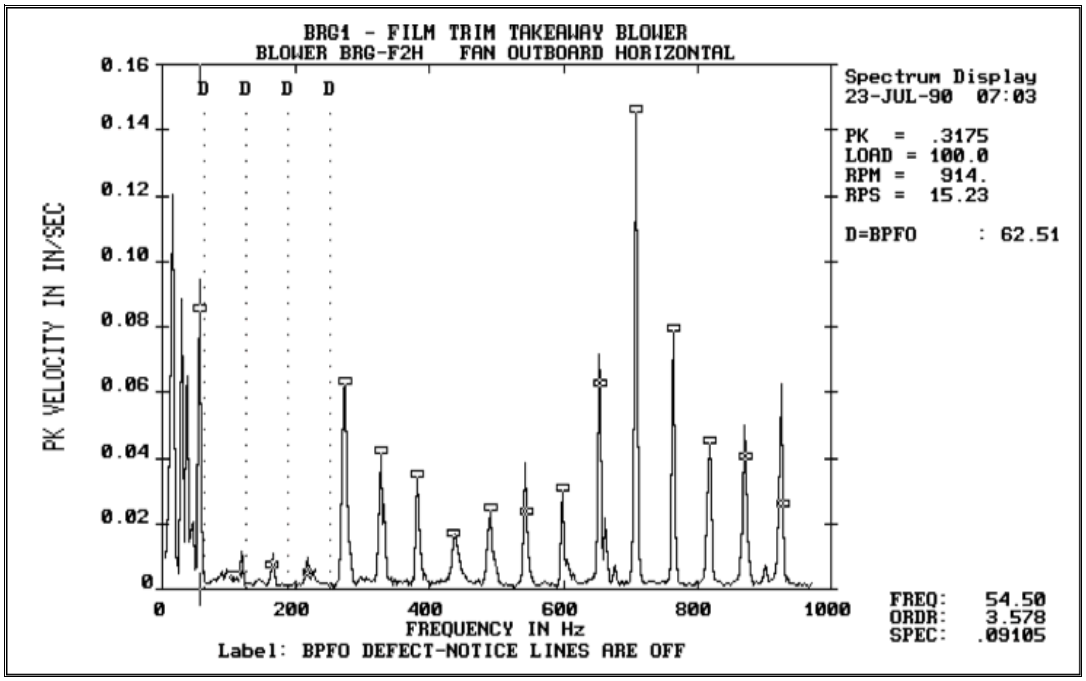


Figure 21

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

There was a significant change from May to July. (Figure 20) The waveform pattern is very repetitive. There is a discrete impacting and ringing down. An approximate 15 G swing, with a Crest Factor of almost 4 is typical of catastrophic bearing failure. (See Figure 22)

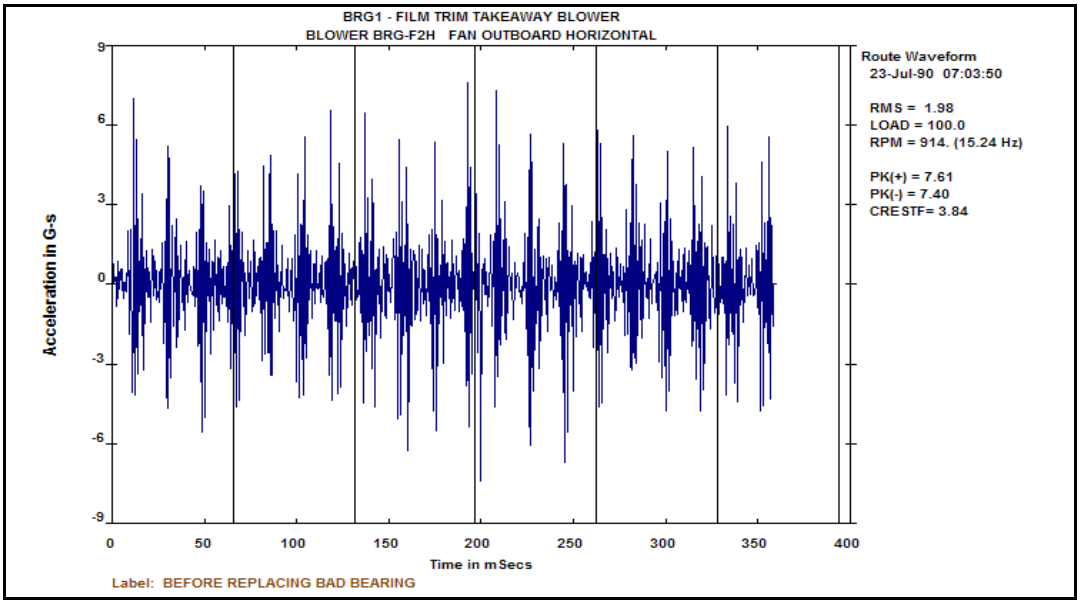


Figure 22

After the bearing was replaced, the high frequency energy was reduced significantly. The turning speed and harmonics are probably coming from the action of the blower. Some trim balancing may be in order. (See Figure 23)

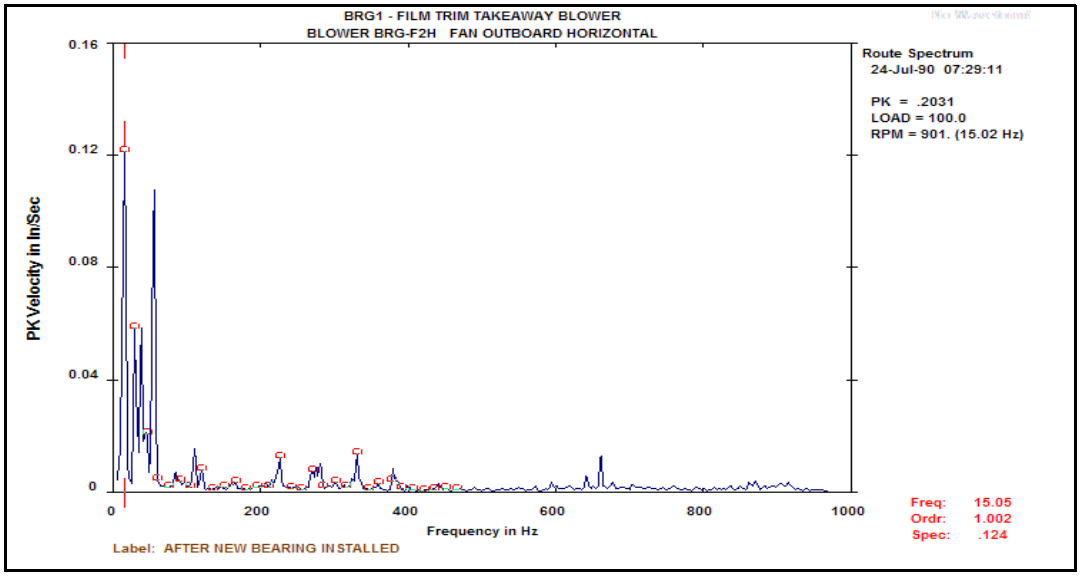


Figure 23

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #5 - Paper Machine Press Roll Bearing

The multiple spectra plot in Figure 24 shows that the harmonics of TS (looseness) and high frequency activity appear on this roller bearing in June.

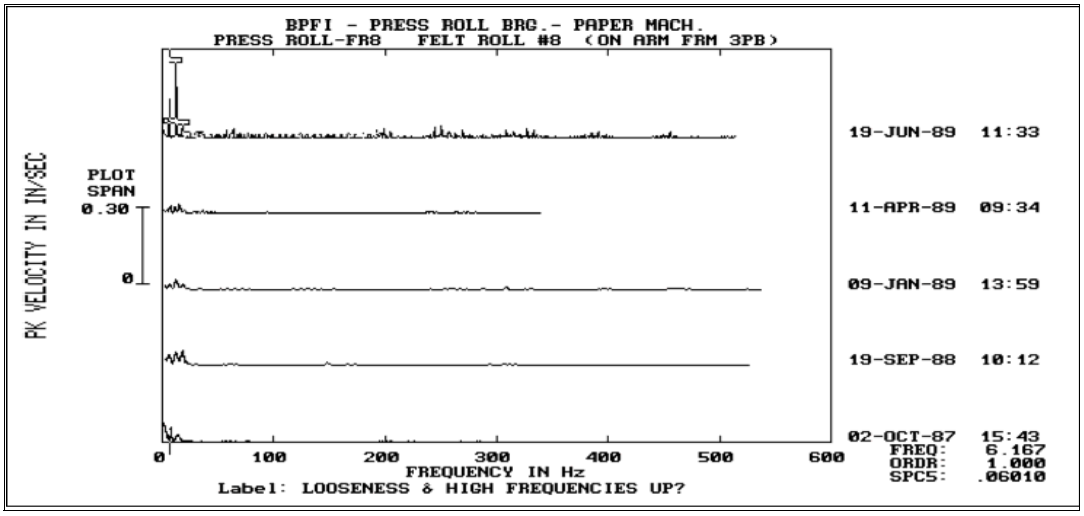


Figure 24

The spectrum in Figure 25 exhibits a pattern typical of a cracked inner race. The mounded groups of peaks represent 1x modulation of the BPFI harmonics. Nonsynchronous peaks are separated by 1 order. The looseness (2X dominant) results from the cracked inner race, creating a loose fit on the shaft.

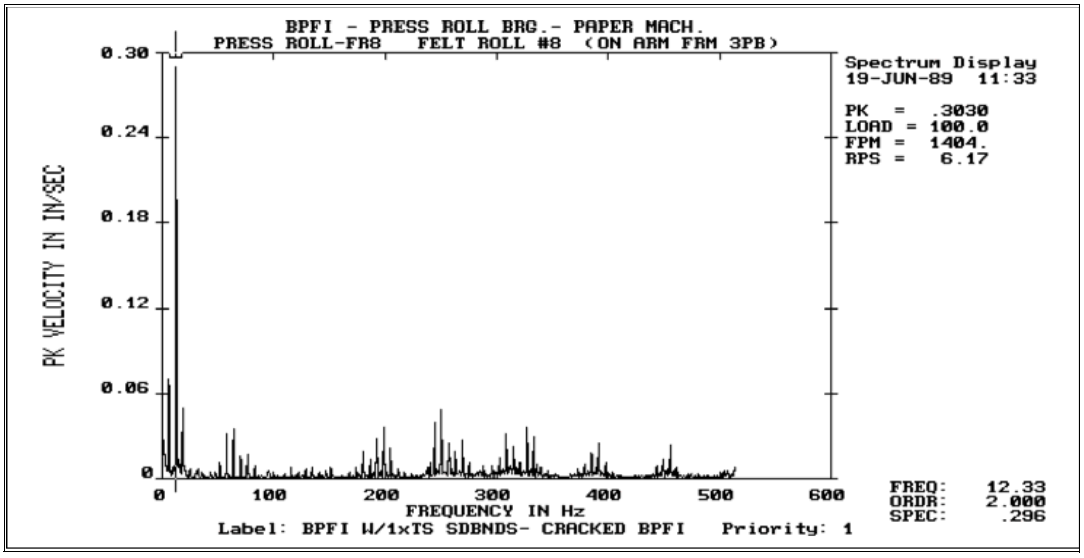


Figure 25

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Peak No.	Frequency (Hz)	Peak Value	Order Value	Peak No.	Frequency (Hz)	Peak Value	Order Value
1	1.14	.02888	.18	**16**	244.35	.0333	39.13
2	2.07	.0160	.33	**17**	250.59	.0416	40.13
3	3.06	.0112	.49	**18**	256.83	.0266	41.13
4	6.24	.0752	1.00	**19**	263.05	.0170	42.13
5	12.48	.2544	2.00	**20**	269.29	.0230	43.13
6	15.01	.0117	2.40	**21**	302.36	.0145	48.42
7	18.71	.0422	3.00	**22**	308.56	.0266	49.42
8	57.97	.0262	9.28	**23**	314.81	.0201	50.42
9	64.21	.0322	10.28	**24**	321.04	.0135	51.41
10	70.45	.0158	11.28	**25**	327.27	.0309	52.41
11	76.69	.0142	12.28	**26**	333.52	.0243	53.41
12	180.15	.0162	28.85	**27**	385.24	.0185	61.70
13	192.62	.0289	30.85	**28**	391.48	.0216	62.70
14	198.85	.0336	31.85	**29**	449.45	.0136	71.98
15	205.09	.0182	32.84	**30**	455.67	.0199	72.98
Total Mag .3053		Subsynchronous .0489 / 3%		Synchronous .2795 / 79%		Nonsynchronous .1328 / 19%	

Table 5

The peak list in Table 5 helps break down the categories of vibration energy. Proportionately, there is more synchronous energy than non-synchronous. A great deal of this synchronous energy is coming from the 2xTS peak from the looseness in the cracked inner race.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Balls or rollers passing over a crack in the race cause a sharp impact and ringing pattern in the waveform. The pattern is quite distinct. The amplitudes, however, are relatively moderate because of the slow roll speed and the low F_{max}. Remember, acceleration tends to accentuate peaks exponentially as frequencies increase.

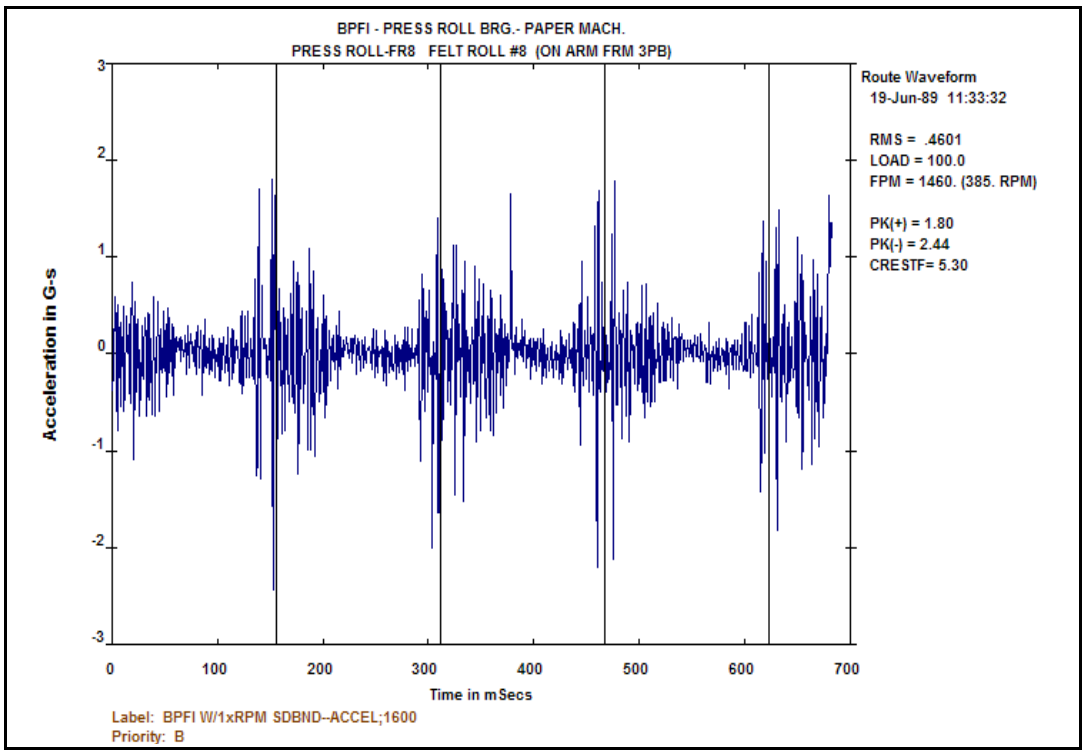


Figure 26

Remember also that BPFI defects often prove more difficult to detect because of their poor transmission path to the transducer. Take seriously the very presence of BPFI defects or patterns when conducting diagnostics - regardless of amplitude.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #6 - Reflux Pump North 2050

On the plot in Figure 27, about 93 Hz separates one peak from the next among the group of seven peaks between 800 and 1600 Hz. This problem is difficult to diagnose. The spacing is about 3.2 orders. A BSF or BPFO is possible. The frequencies do not match, but significant energy exists in the waveform. (See Figure 28)

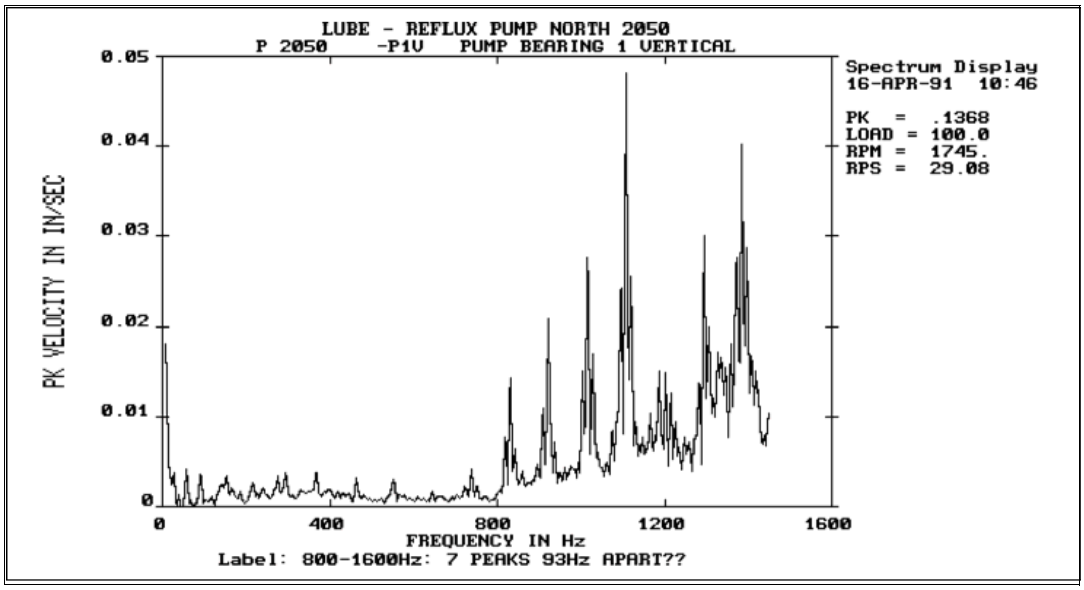


Figure 27

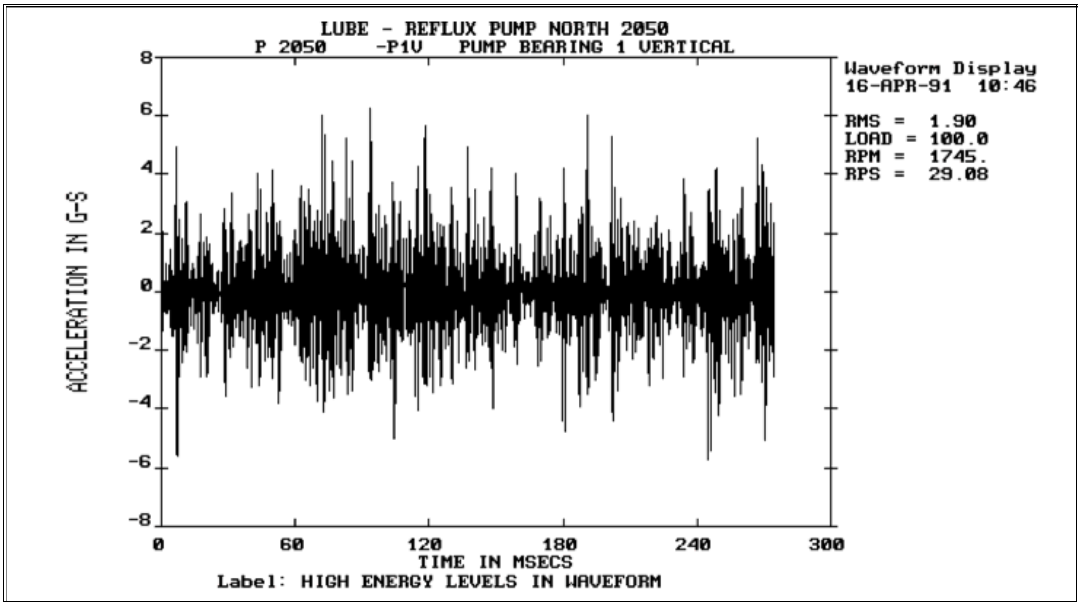


Figure 28

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Loss of lubrication is sometimes difficult to detect before catastrophic failure because the bearing usually fails before the frequencies are seen in vibration data. Typically the dry bearing generates a pattern similar to the one illustrated in Figure 29.

A group of peaks appears in the 800- to 1600-Hz range separated by 80 and 130 Hz (in this case, 93 Hz).

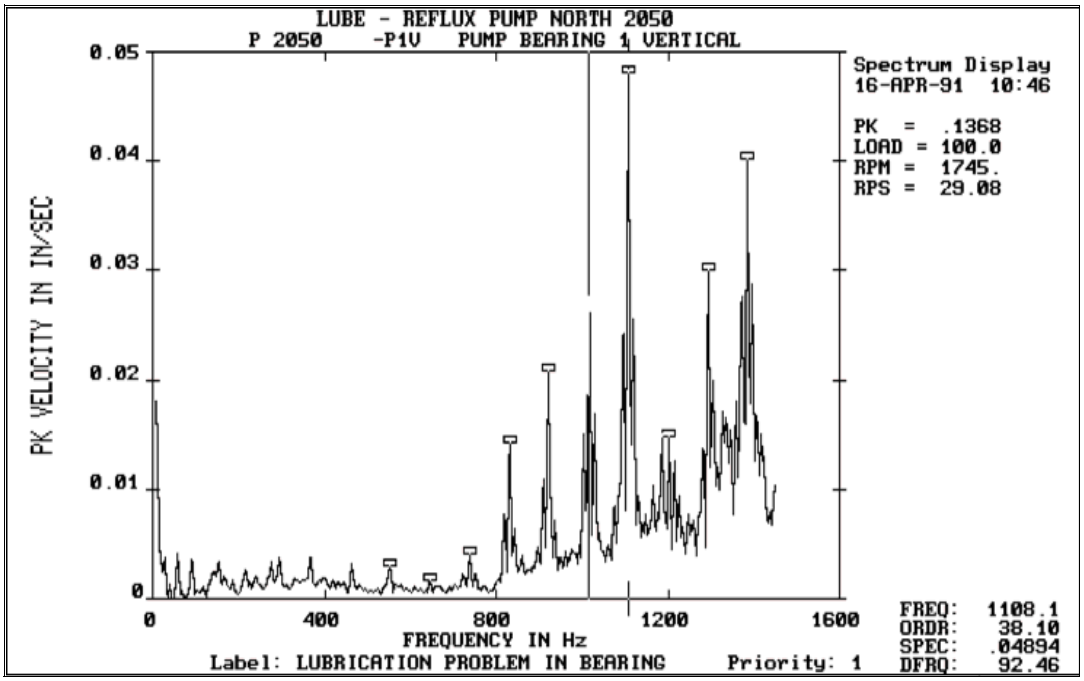


Figure 29

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #7 - Fan Pump

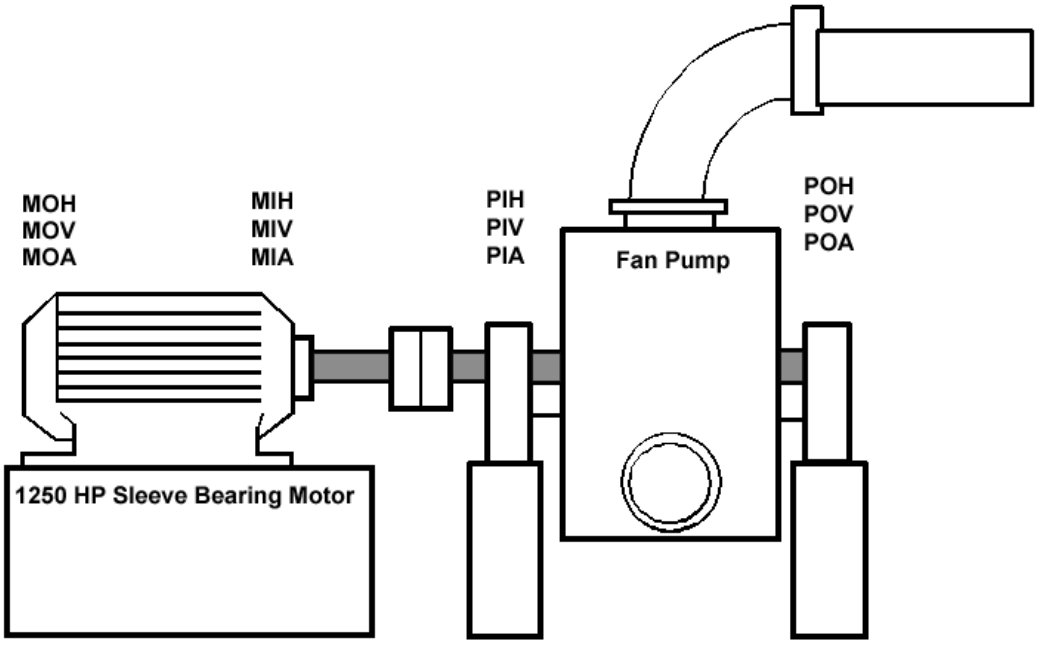


Figure 30

The location of the measurement points appear in the diagram in Figure 30. Access to the inboard end of the motor is prevented by the coupling guard.

◆ The motor was replaced in an effort to reduce vibration levels. The unit remains noisy.
◆ The waveform verified impacting.
◆ Suspect roller bearing degradation.
◆ Spectral data indicates looseness is present.
◆ Spherical roller bearings show higher axial readings than cylindrical bearings.
◆ Spherical bearings also show defects at higher frequencies.

Additional information is given with Figure 31.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The multi-point plot in Figure 31 shows data for the motor outboard points. All the levels appear very low in amplitude, but note the location of the dominant peaks. The 2xTS peaks equal or exceed the amplitude of the other peaks in the spectra. This probably means that the motor is slightly misaligned to the pump. The low vibration levels mean that immediate correction is probably unnecessary.

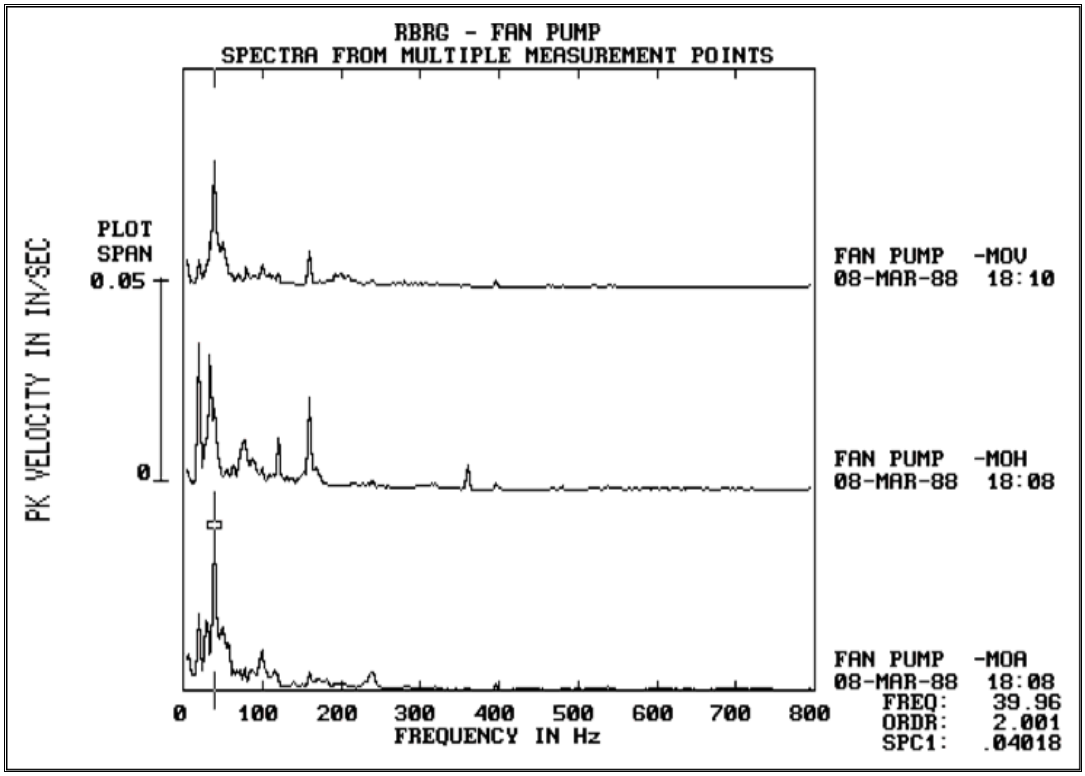


Figure 31

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Data from the pump points shows broad-banded humps of energy along with some low frequency peaks. The humps of energy range from 200 to 400 Hz (12000 to 24000 RPM, or about 10 to 20 orders of turning speed). (See Figure 32)

Examine the low frequency peaks more closely. They might be harmonics of turning speed, or they could be harmonics of some bearing frequency. The full scale range of all the plots is low at 0.1 IPS, but the broad humps generate quite a bit of energy. (See Figure 32)

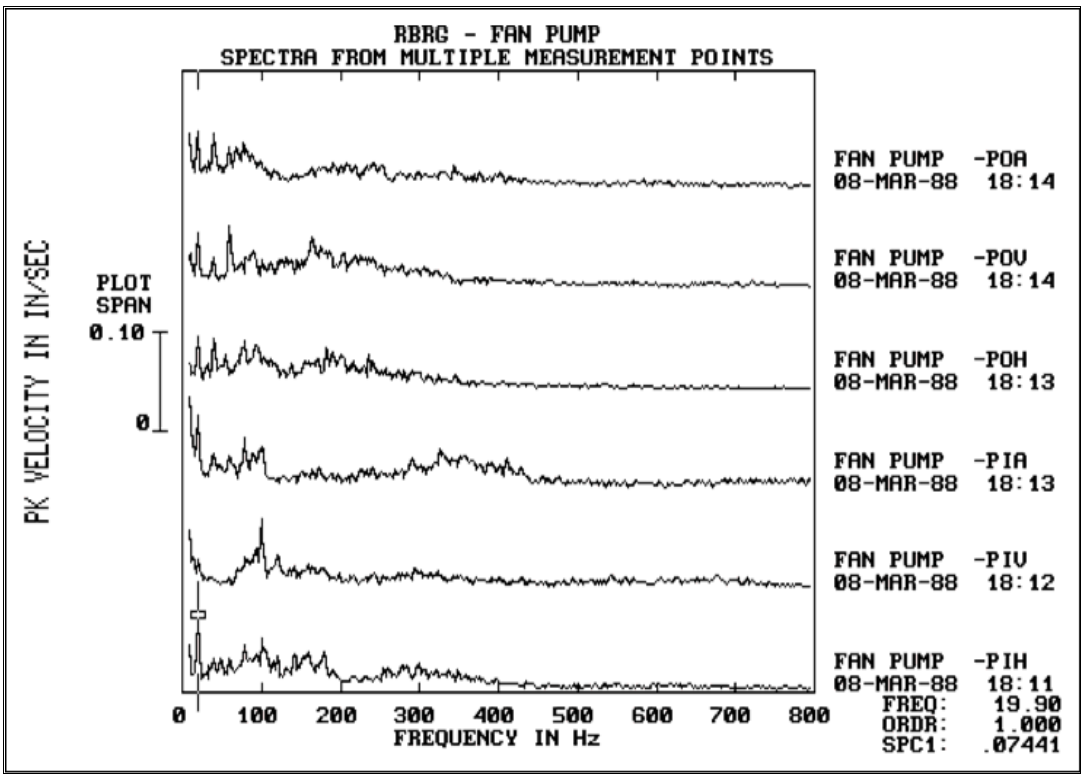


Figure 32

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

A full-screen plot of the Pump Inboard Horizontal (PIH) appears in Figure 33. The Fault Frequency lines mark harmonics of turning speed. Harmonics of nonsynchronous peaks are also present. The harmonic cursor highlights one such peak at 4.763xTS. Note that no individual peak has a very high amplitude.

The full-scale range of the plot is only 0.1 IPS, but the overall value of the measurement point is almost 0.3 IPS. Broad humps of energy cause the relatively high overall value. Random impacts (shown in the time waveform in Figure 34) tend to generate broad humps in the spectrum. The calculations that create the spectrum from the time waveform cannot translate the impacts to any specific frequency, which causes the appearance of the humps instead of a peak.

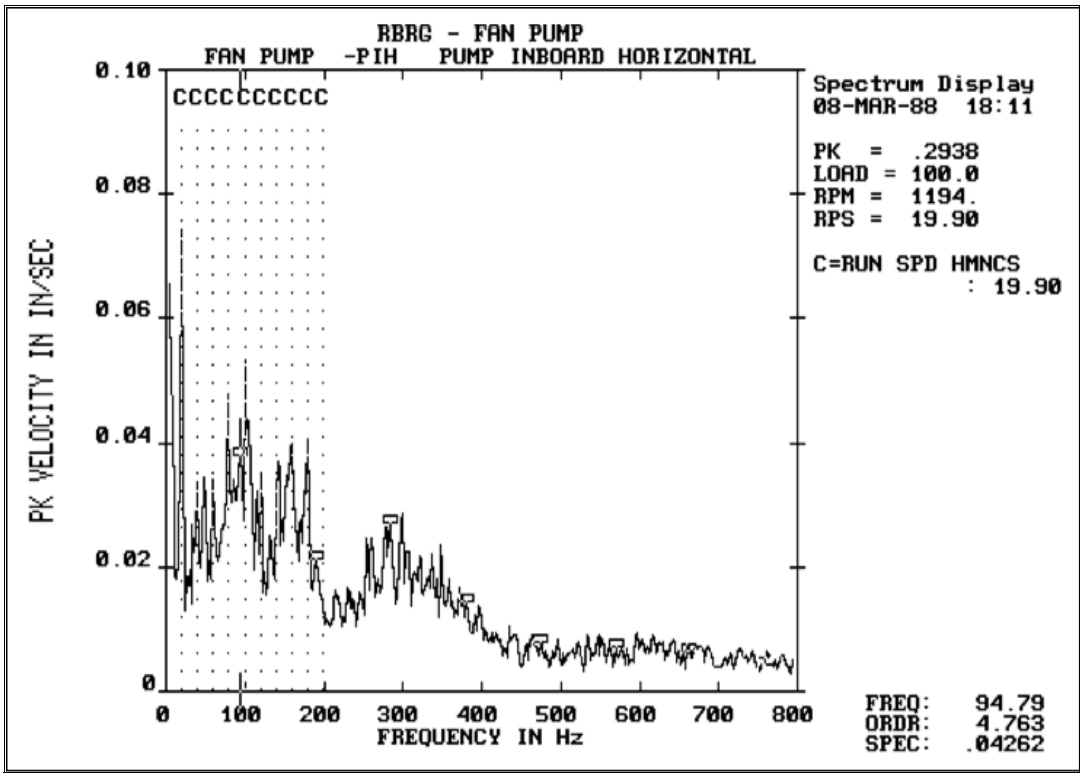


Figure 33

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The number and amplitude of the spikes in the time waveform in Figure 34 confirm the presence of severe impacting. Many impacts exceed ± 2 g's in amplitude. These impact levels generate large amounts of energy in this massive pump. The bearings in turn absorb this impact energy, which damages them rapidly. The random and complex pattern of the waveform proves impossible to transform into a clean spectrum, resulting in the broad humps of energy shown in Figure 33.

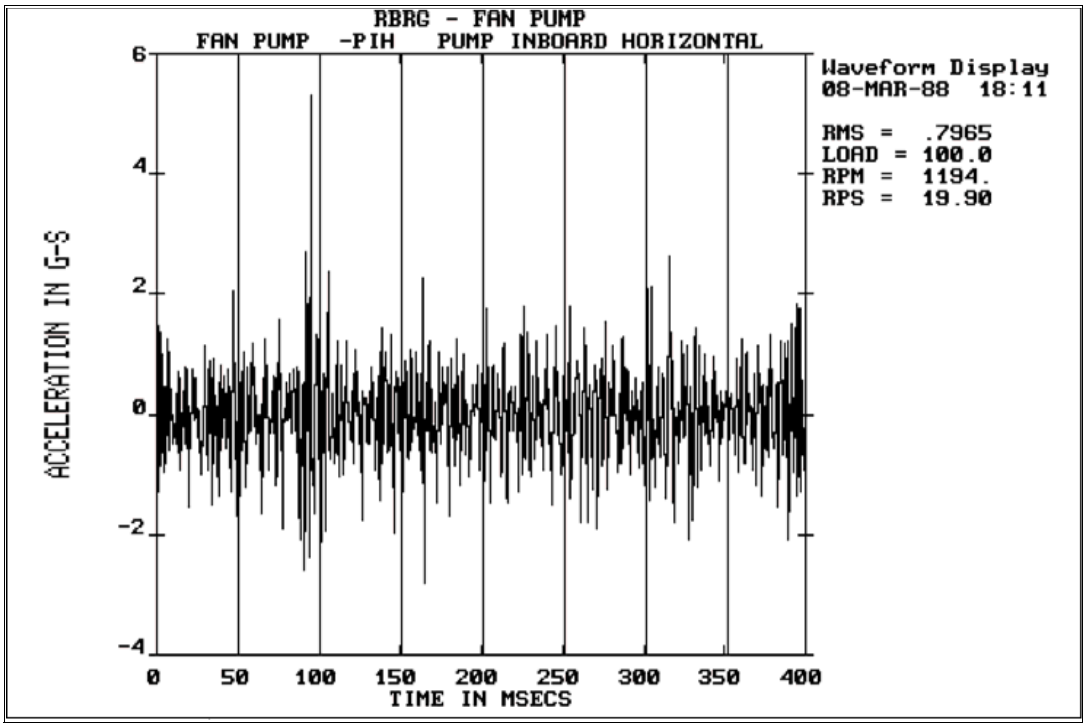


Figure 34

Sometimes the real difficulty lies in deciding which machine is suffering the most from excessive vibration, thereby knowing where to focus resources. Consideration must be given to the specific frequency or range where the vibration is originating, and how the vibration is distributed. For example, a machine may be able to operate normally at 0.125 inches per second at lower frequencies at or around turning speed. However, that same machine will fail if that same amplitude is reached at higher frequencies such as where rolling element bearings are manifest.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History # 8

The following case histories demonstrate the need to consider parameters other than just overall values, or depending on amplitudes alone, as the sole data for assessing the condition of the machine.

The spectrum in Figure 35 illustrates why a single overall value is not recommended to determine the condition of a machine. This spectrum comes from a pump with a rolling element bearing. It has an overall value of just under 0.3 IPS. No one specific peak is very high. However, broad-banded energy dominates the plot, indicating a severe bearing defect. If the analyst relied on amplitude alone to reach a conclusion, a significant problem may be missed.

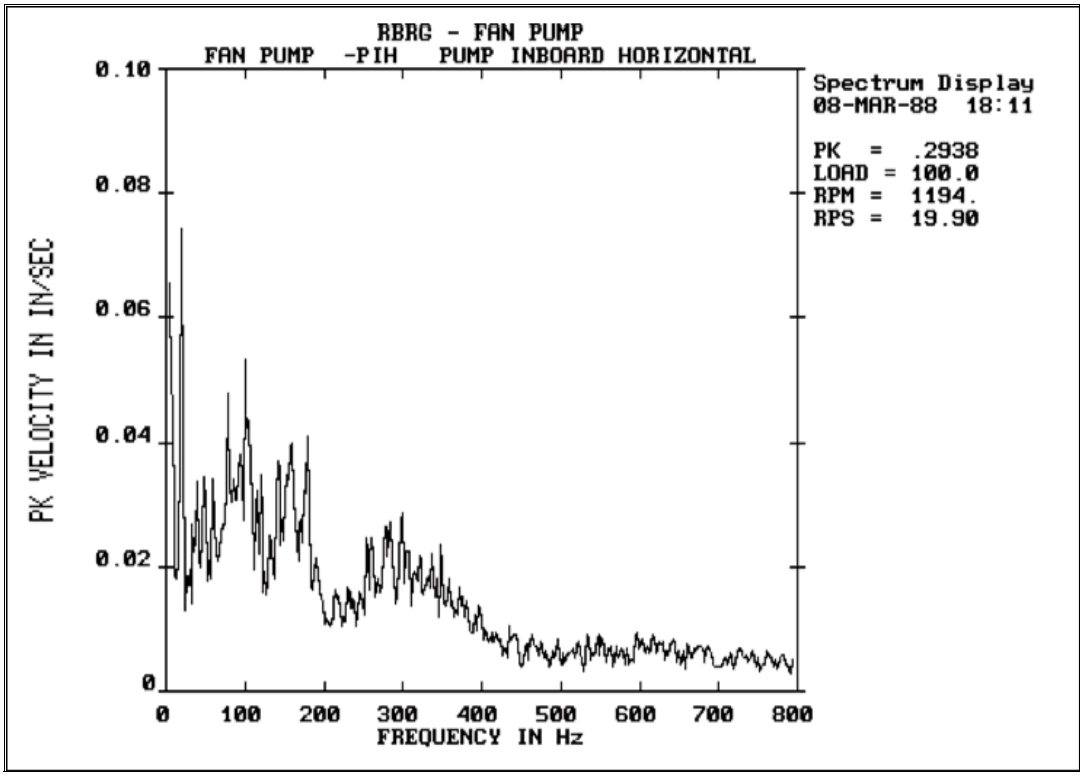


Figure 35

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The spectrum in Figure 36 comes from a steam turbine with an apparent unbalance problem. Like the plot in Figure 35, this spectrum shows an overall value of about 0.3 IPS. Virtually all of the energy of the spectrum comes from the single peak at 1xTS. Compare the amplitude of the single peak in the spectrum to the overall value of the spectrum.

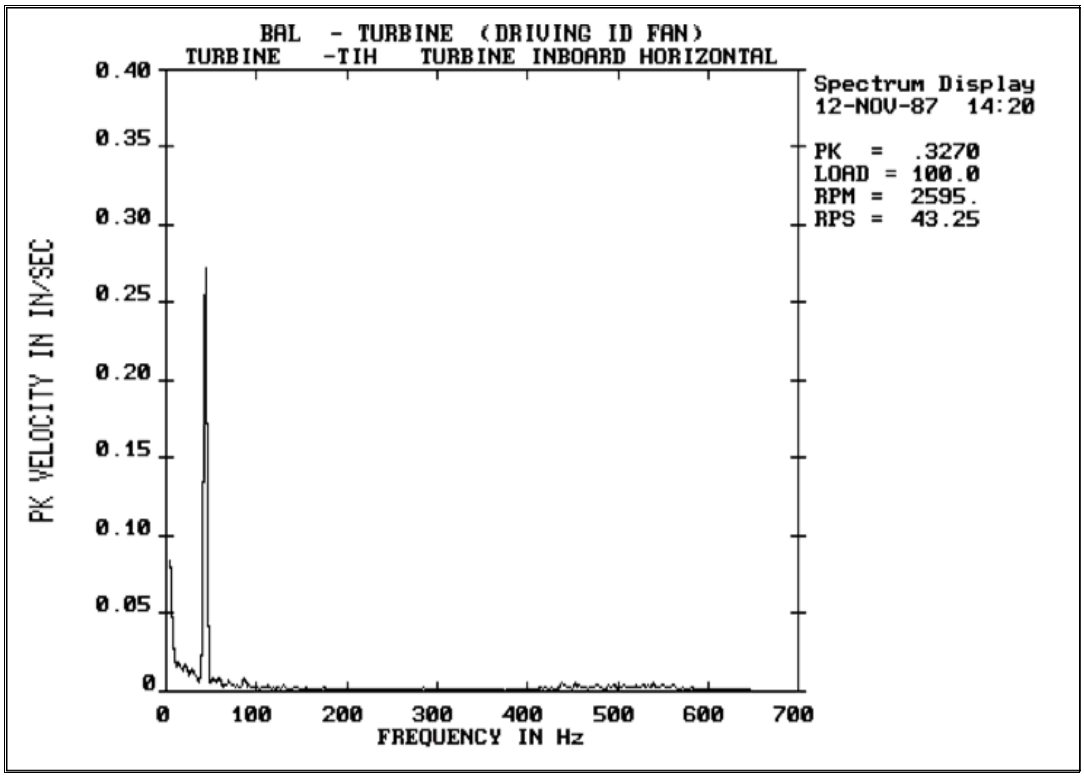


Figure 36

Compare the respective time waveforms from the turbine and the fan pump to help determine which machine has the worse problem.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The waveform in Figure 37 comes from the pump bearing. It shows high levels of impacts. One impact reaches almost 5 g's, and most are in the ± 2 g range. The RMS overall value of the waveform approaches 0.80 g's.

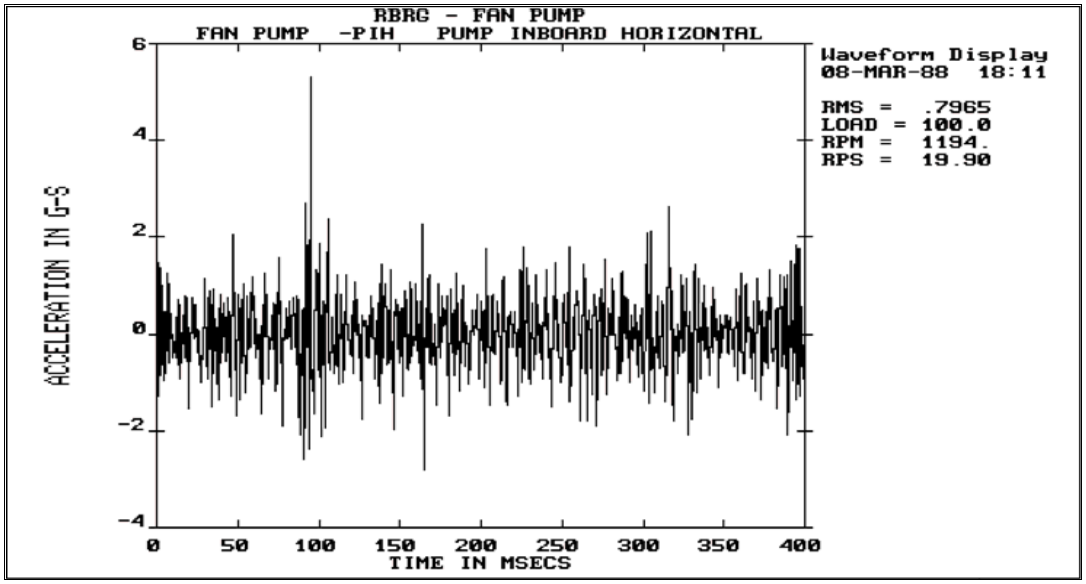


Figure 37

The waveform in Figure 38 is from the steam turbine. There is virtually no impacting, with no peaks exceeding ± 1g, with approximately 0.23g's RMS. The sinusoidal nature translates well into a spectrum. Even though the pump waveform's RMS was more than triple that of the turbine, the ease of FFT from the turbine waveform results in a higher spectral value.

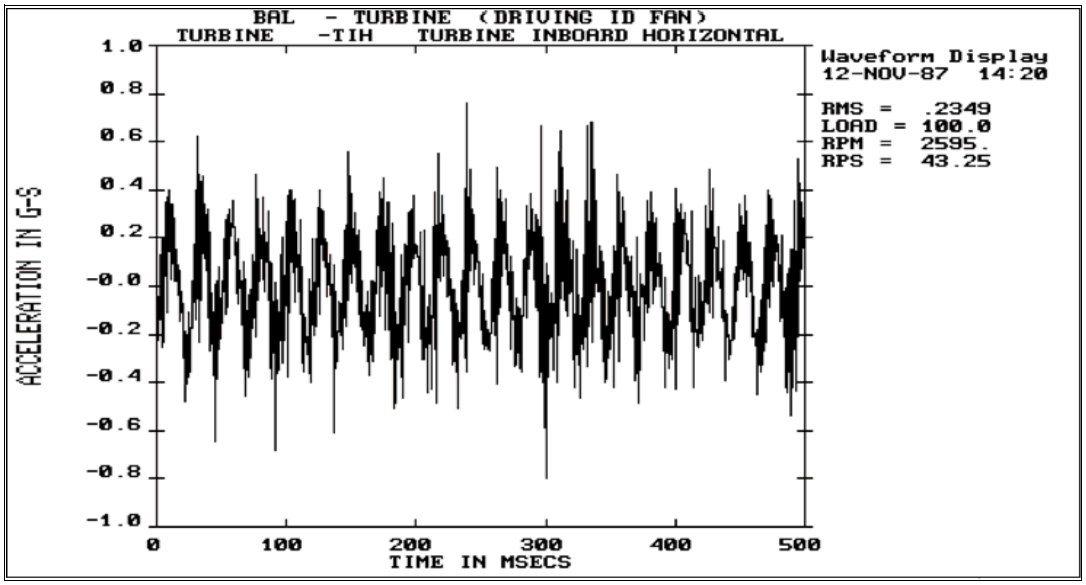


Figure 38

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

To help determine which bearing needs attention, compare the data from multiple points and directions. Figures 39, 40, and 41 were taken from the inboard end of the pump.

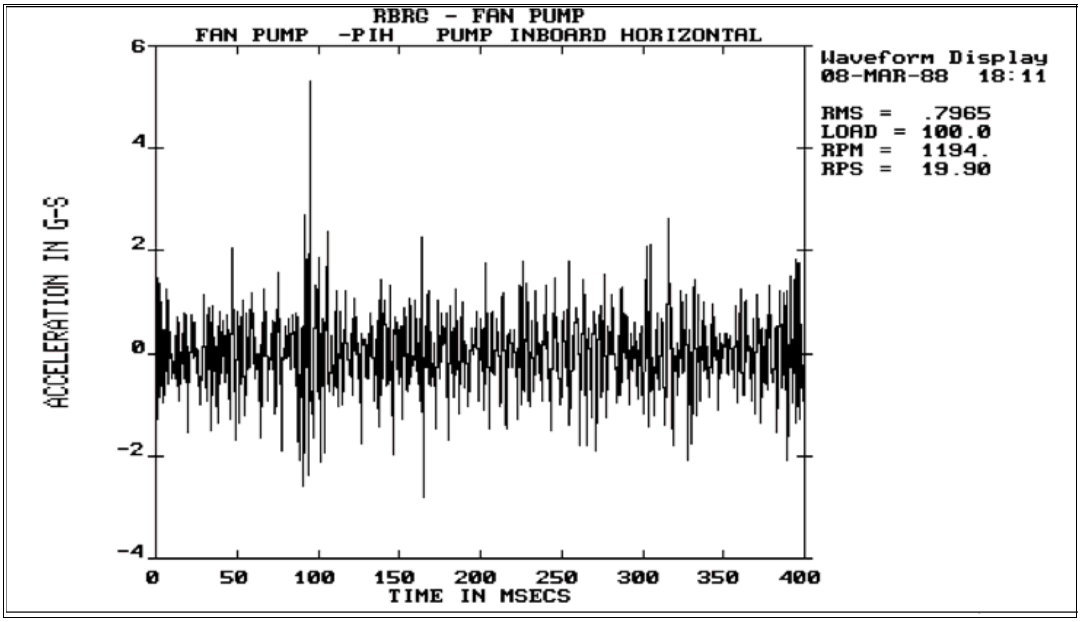


Figure 39

Of the two, horizontal and vertical, the horizontal point is suffering more impacting of over 5g's. However, the vertical direction seems to contain more overall energy.

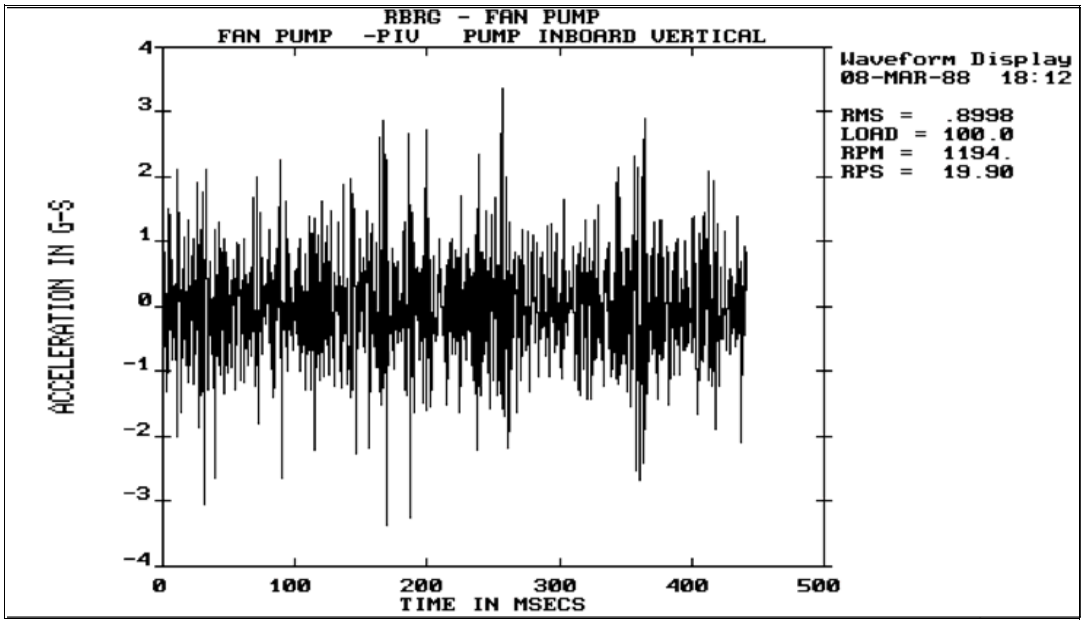


Figure 40

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Higher impacts and a large **g swing**, usually indicates a bearing with more and larger defects. After comparing the data from all three directions, it is apparent that the axial point is more energetic than horizontal or vertical. (See Figure 41)

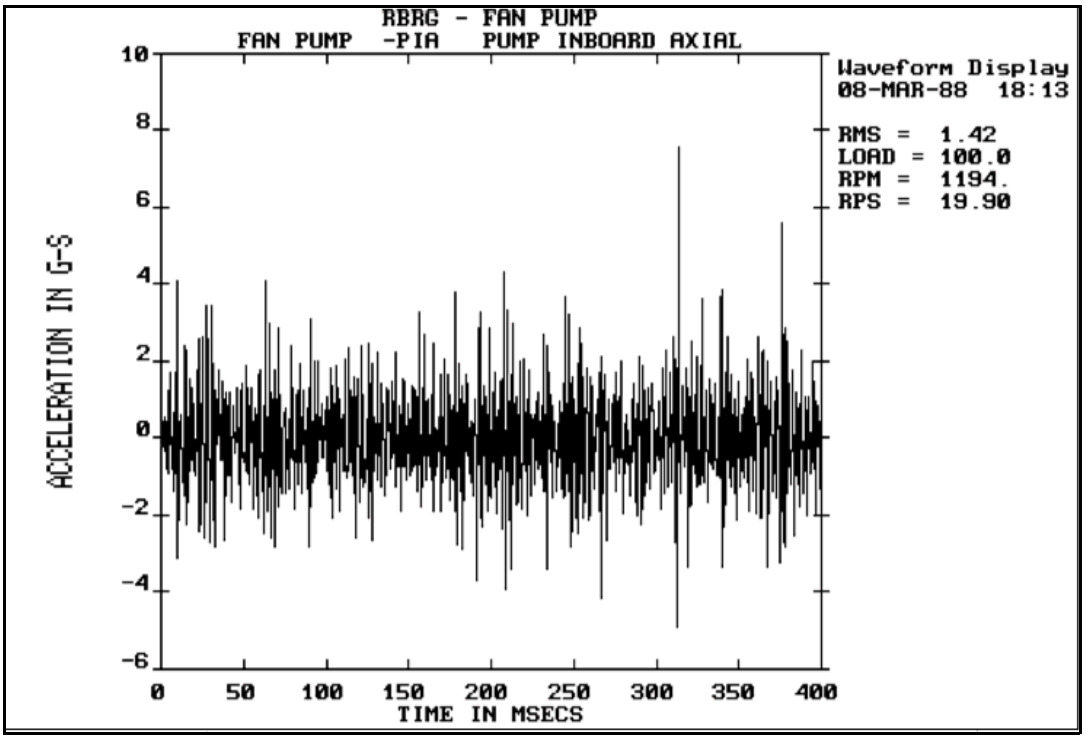


Figure 41

Point	RMS	Point	RMS
PIH =	0.80 g's	POH =	0.48 g's
PIV =	0.89 g's	POV =	0.77 g's
PIA =	1.4 g's	POA =	0.83 g's

Table 6

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Comparing the Inboard data to the Outboard will help determine which bearing should be changed. See Figures 42 and 43 for the Outboard waveform signatures.

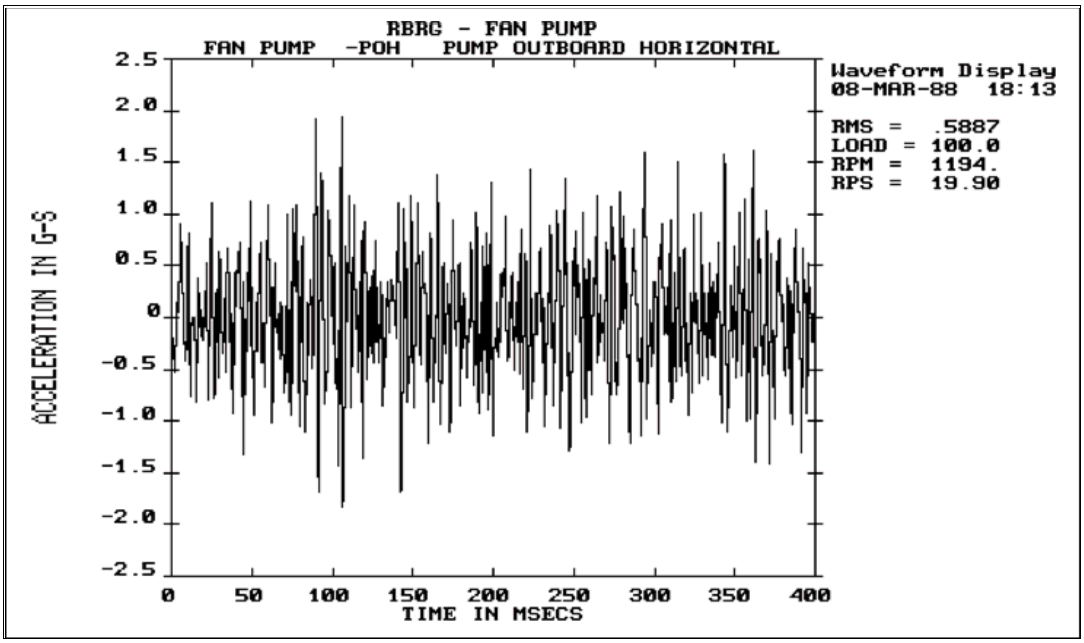


Figure 42

The Vertical data seems to be more energetic than the Horizontal. It also has a higher g swing and RMS.

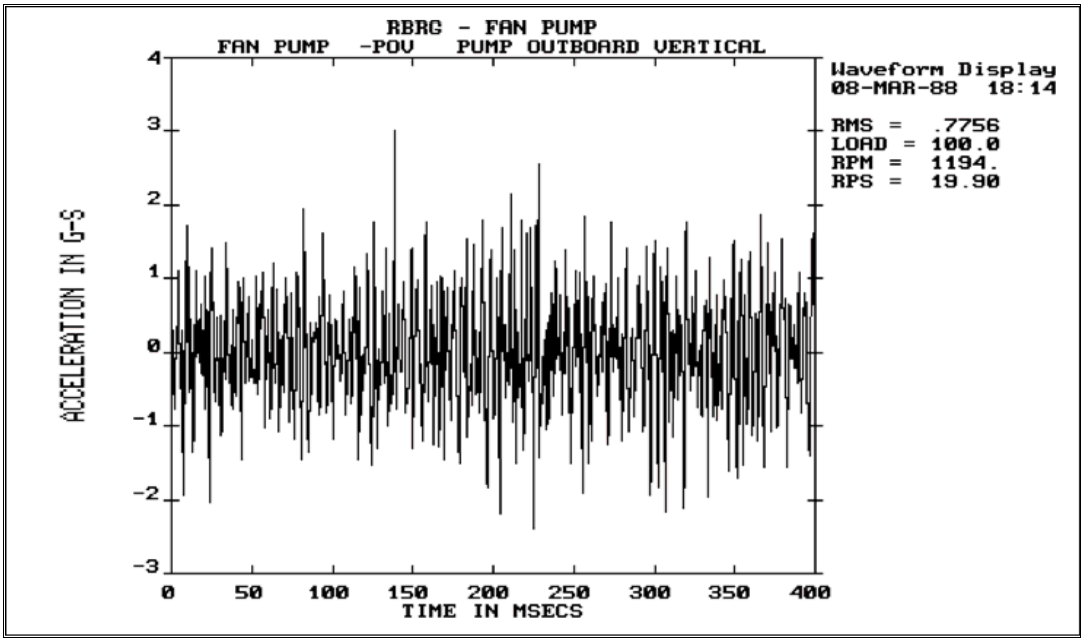


Figure 43

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

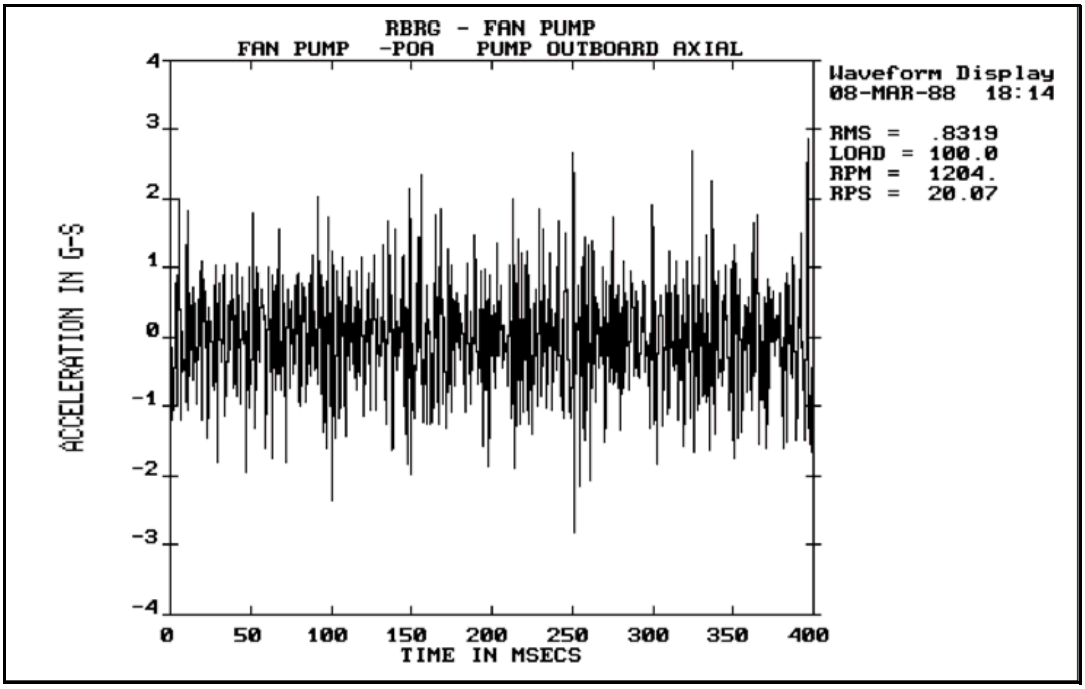


Figure 44

Point	RMS	Point	RMS
PIH =	0.80 g's	POH =	0.48 g's
PIV =	0.89 g's	POV =	0.77 g's
PIA =	1.4 g's	POA =	0.83 g's

Table 6

The axial point exhibits more energy. (See Figure 44) The RMS levels for the outboard points are consistently lower than those for the inboard points. More importantly, the scale of the plots shows the level of the spikes or impacts much higher on the inboard points. Therefore, the inboard bearing probably has the most severe defects. While both bearings are probably in need of attention, if only one could be changed, the inboard would be the most likely candidate.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #9 - Paper Machine Dryer Roll

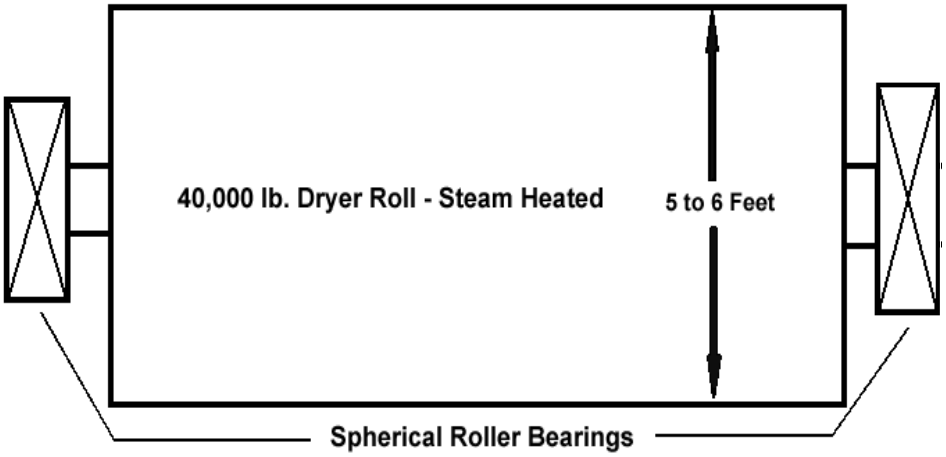


Figure 45

◆ Bearings are pressure lubricated from a common sump.
◆ The surface temperature of the roll must remain above 212° F to prevent rust formation. Since the roll expands under this heat, one of the bearings must be free to float in the axial direction. The floating bearing is usually the one opposite the gear end of the roll.
◆ The helical gear driving the roll produces an axial load on the bearings. This means that axial measurements are usually the best way to find bearing defects.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The spectrum in Figure 46 shows five harmonics of the ball pass frequency on the outer race. Some broadband energy also appears in the plot.

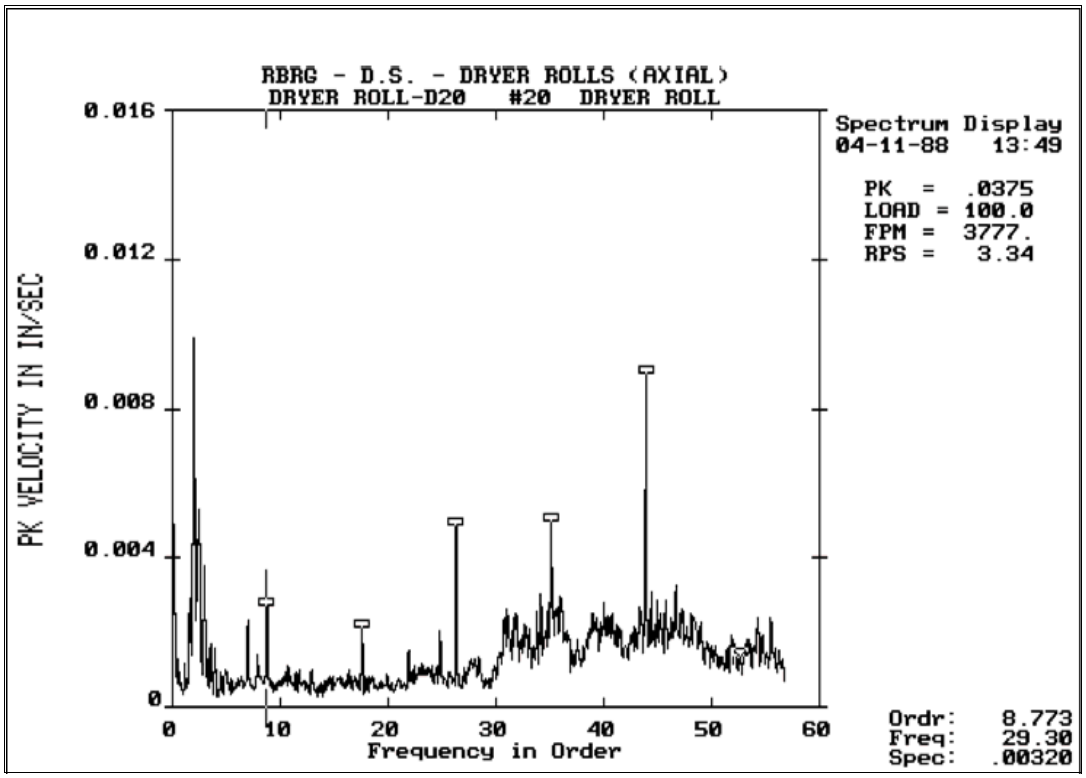


Figure 46

Take very seriously the presence of apparent low amplitude, high frequency peaks, especially on slower speed equipment. This roll rotates at only a little more than 3 Hz or 180 rpm.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The multiple spectra plot in Figure 47 shows data from April through August. The bearing on the opposite end of the roll is free to float axially, so peaks at the lower harmonics of run speed (1, 2, 3, and 4xTS) tend to rise and fall as the machine speed varies.

Although the machine speed has varied, the amplitudes of the bearing defect frequencies have not changed significantly in this time. Therefore, the defect is not growing rapidly, but it may not rest in the load zone of the bearing. This bearing warrants close attention for signs of any sudden change.

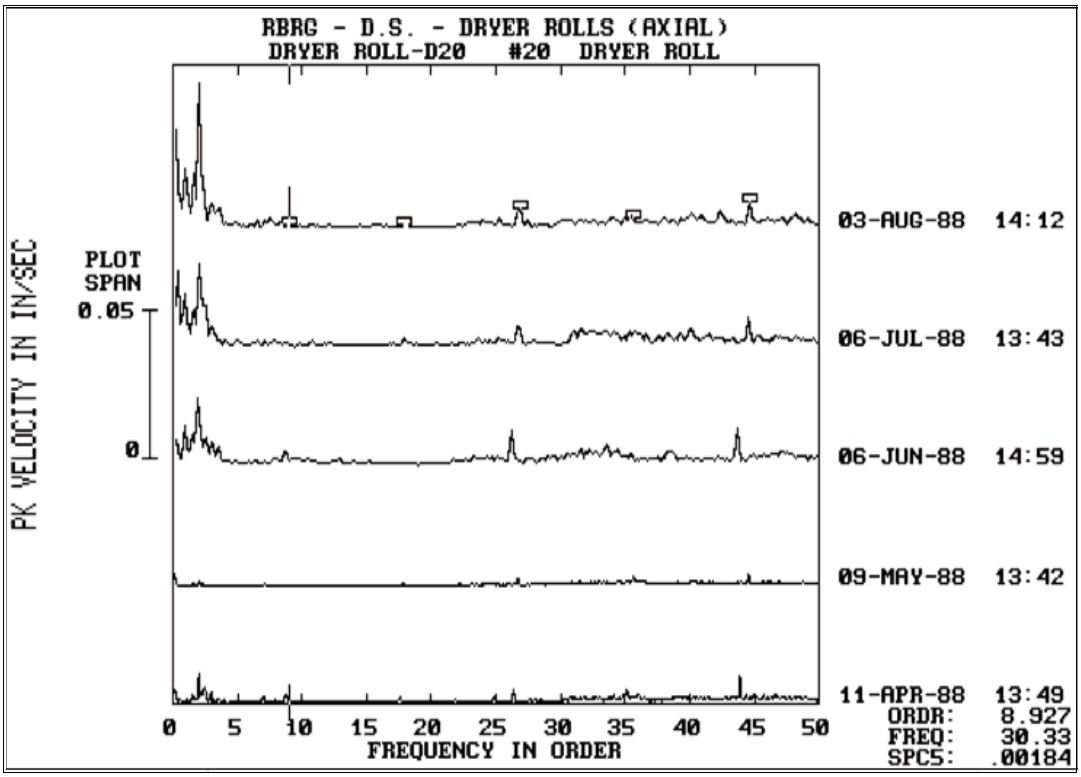


Figure 47

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The multiple spectra plot in Figure 48 comes from another dryer roll of similar design. However, its bearing problem differs significantly. The spectra from April to July 1988 show the sudden appearance of a family of harmonic peaks. The first harmonic of this family occurs at 8.161xTS.

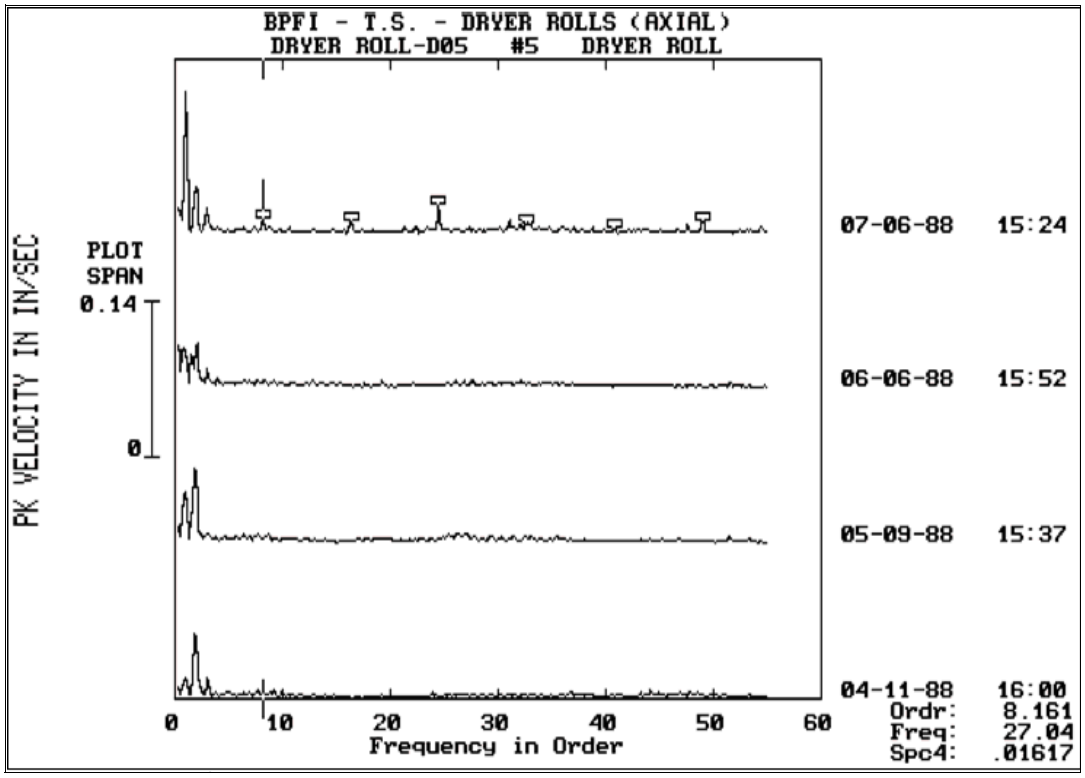


Figure 48

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #10 - Paper Machine Wire Return Roll

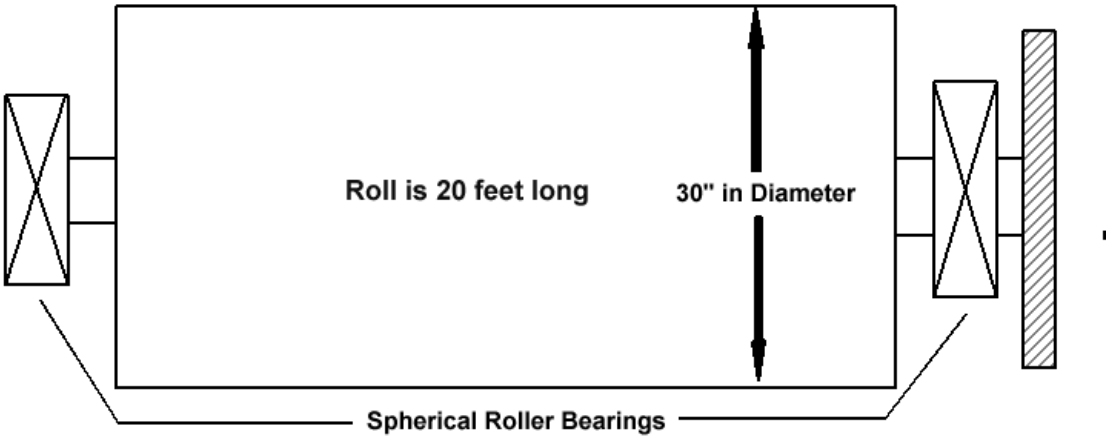


Figure 49

◆ Bearings are pressure lubricated from a common sump.
◆ This roll reverses the direction of the wire on which paper fiber is sprayed to form a paper sheet.
◆ The roll also adds tension to the wire.
◆ No heat is required on this roll.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The plot in Figure 50 comes from axial data on a defective spherical roller bearing. The bearing frequency patterns indicate a problem in spite of the low amplitude. Many peaks occur on the upper and lower sides of 4 and 5xBPFO [Ball Pass Frequency Outer (race defect frequency)]. These peaks are spaced in frequency by 1xTS of the shaft. Therefore, 1xTS sidebands modulate peaks around 4xBPFO and 5xBPFO. See dialog on page 54.

This characteristic indicates that the shaft creates these frequencies. This bearing also warrants close attention to watch for further degradation.

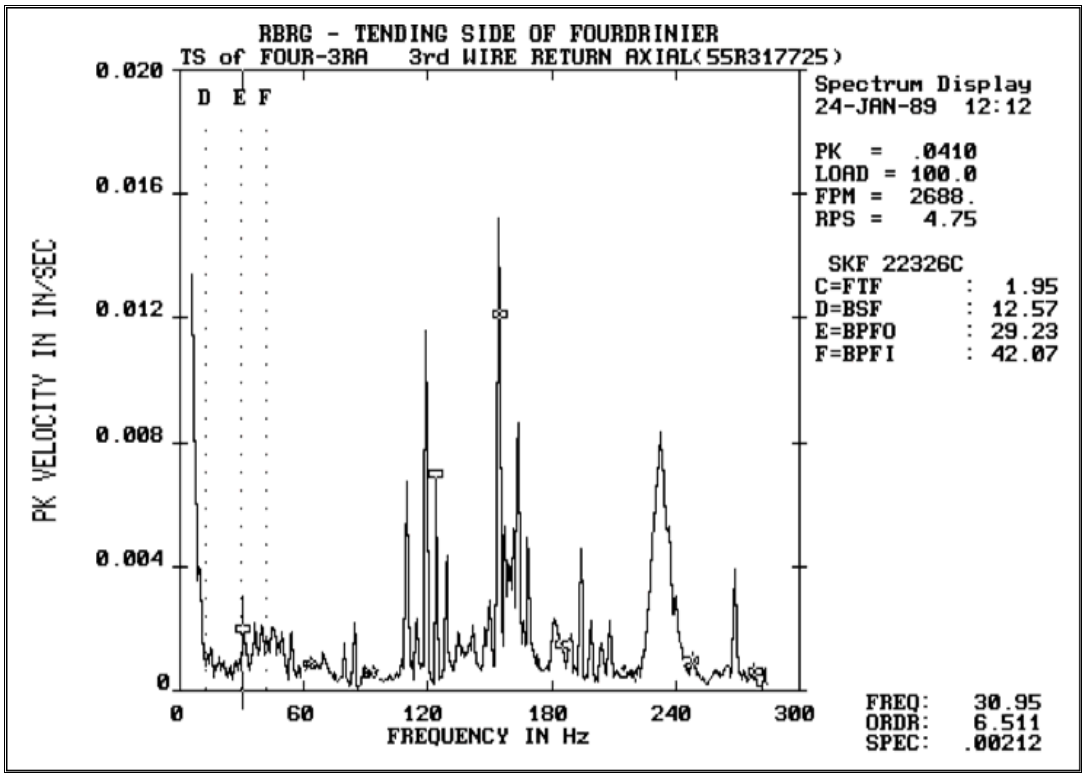


Figure 50

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The primary cursor in Figure 51 is marking the primary BPFI [Ball Pass Frequency Inner (race defect frequency)]. The major peaks seem to appear at 3 and 4xBPFI, which makes it relatively difficult to determine whether this is an inner or outer race problem.

These harmonic peaks of BPFI could be sidebands modulated by the turning speed of the shaft. An inner race defect is potentially more serious than an outer race defect, so the problem requires frequent examination. A time waveform is valuable to help analyze the problem.

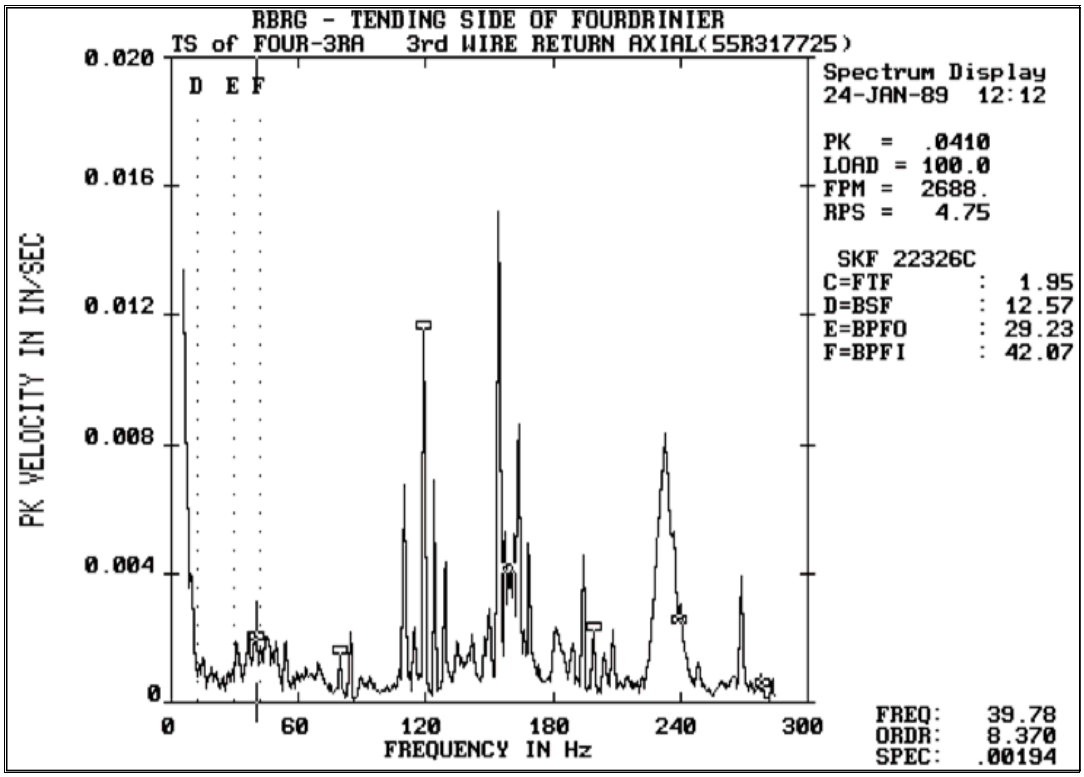


Figure 51

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Figure 52 compares radial and axial readings on this spherical roller bearing. The axial load on this bearing, and the closer proximity of the probe to the bearing in the axial direction, causes the higher axial amplitudes.

This example validates the need for axial measurements on spherical roller bearings. In general, make an axial measurement on all bearings that carry a significant thrust load.

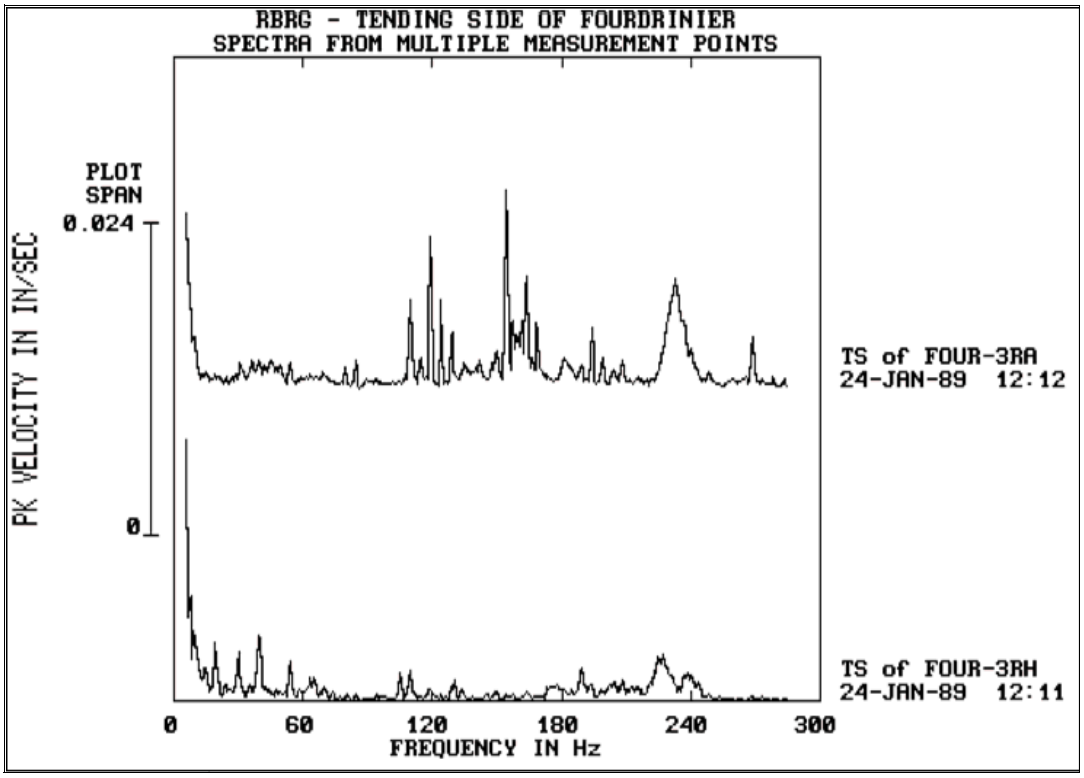


Figure 52

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #11

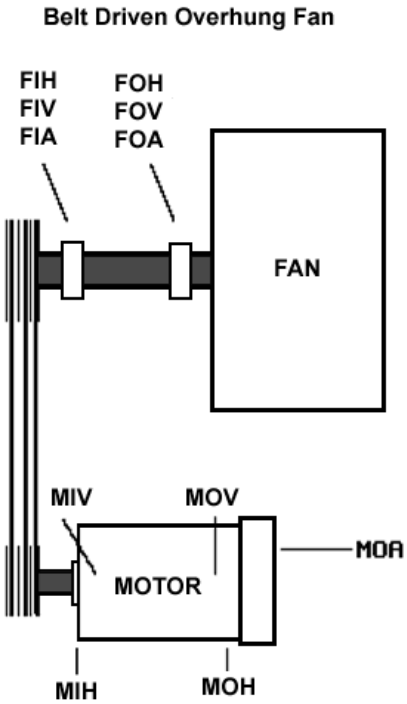


Figure 53

◆ This equipment is located on the roof of the building.
◆ The motor grease fittings have rusted. The fan has no grease fittings.
◆ A common I-beam frame supports the motor and fan.
◆ Although the entire structure vibrates, the vibration amplitudes are much higher on the motor than the fan.
◆ The fan conveys a granular product to the top floor of the building. If the fan fails, the entire line must shut down, because no on-line spare exists.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

According to the multiple spectra plot in Figure 54, the motor is more energetic than the fan. The peaks on the motor spectra appear to be harmonics of 1xTS.

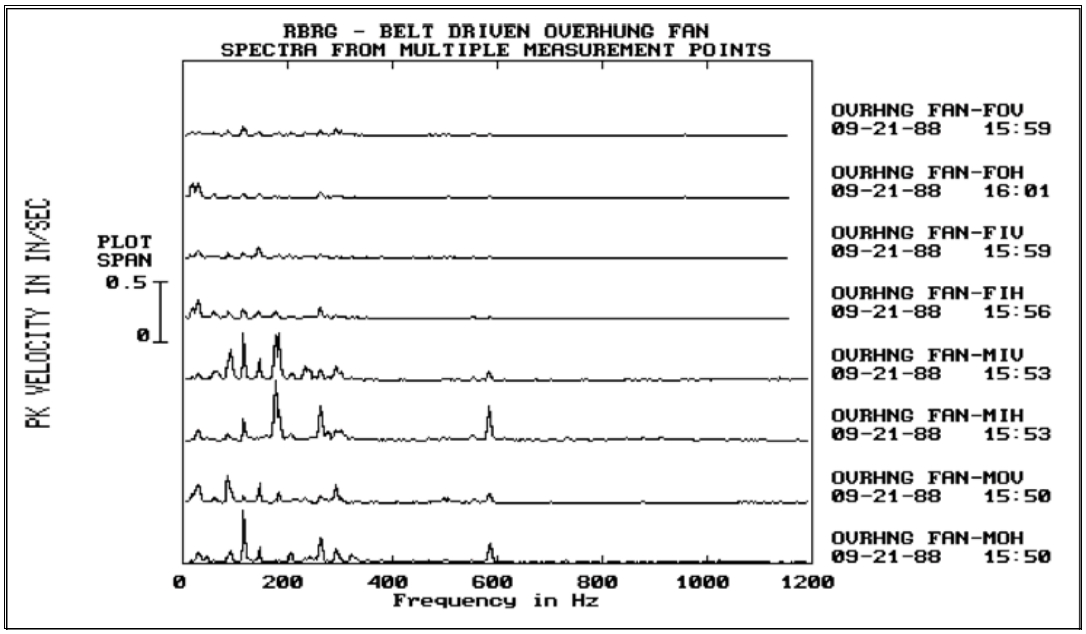


Figure 54

A closer view of the Motor Inboard Vertical (MIV) position indicates the presence of the family of peaks at harmonics of 3.165xTS. Harmonic peaks of 1xTS also appear. (See Figure 55)

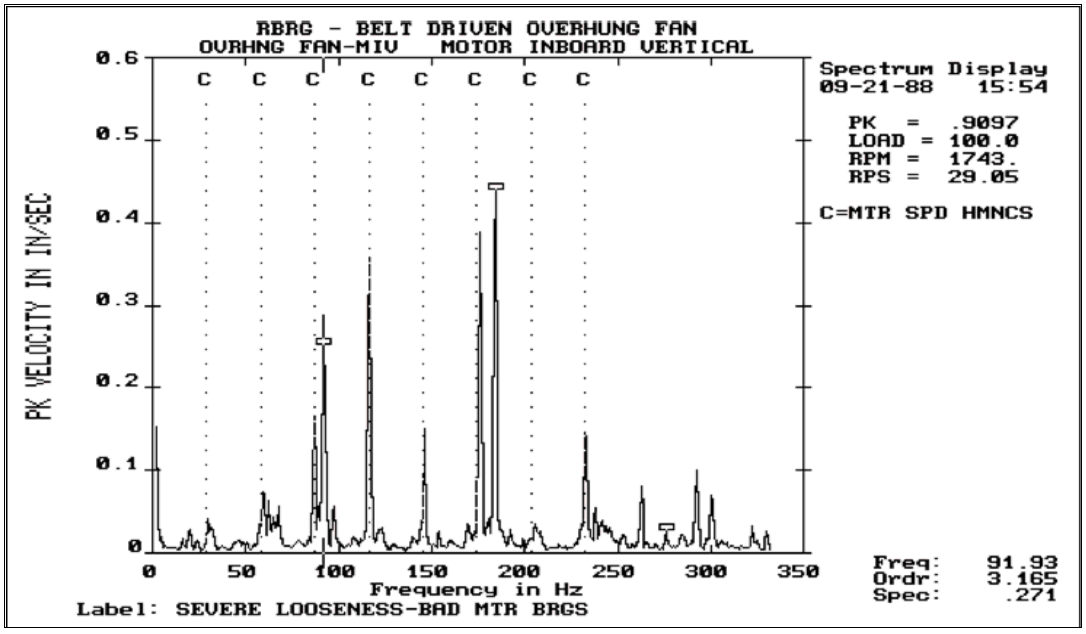


Figure 55

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The time waveform of the motor inboard vertical data shows significant impacting, especially for a velocity waveform. (Figure 56)

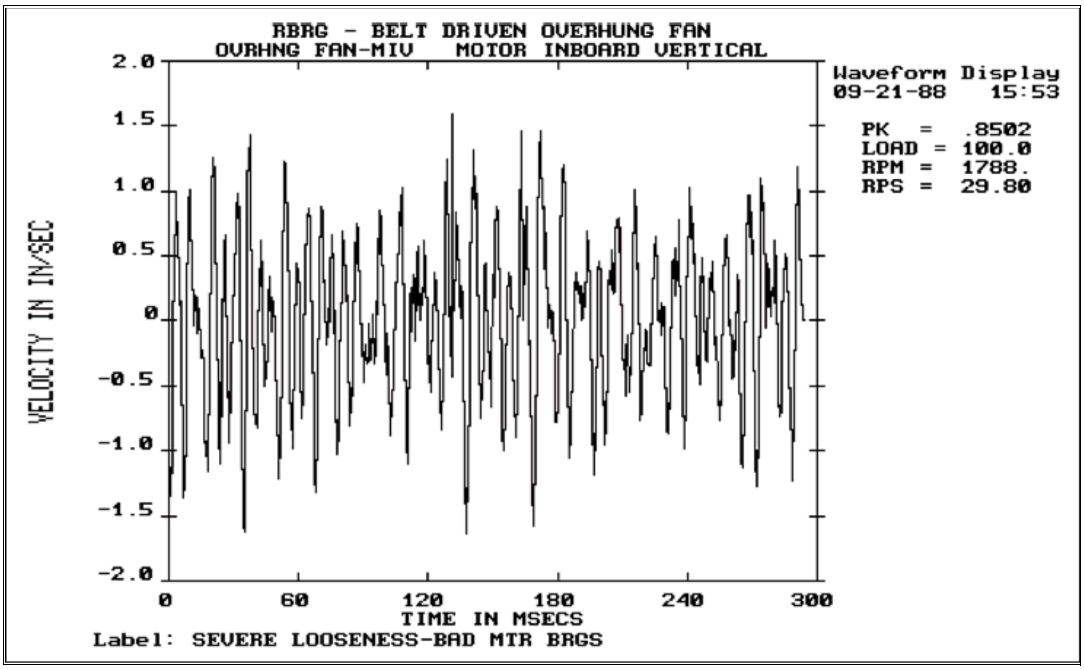


Figure 56

After replacing the motor bearings, virtually all non-synchronous harmonic vibration disappears. The 1xTS peak on MOV increased, however, which indicates a possible belt, sheave, or alignment problem. (Figure 57)

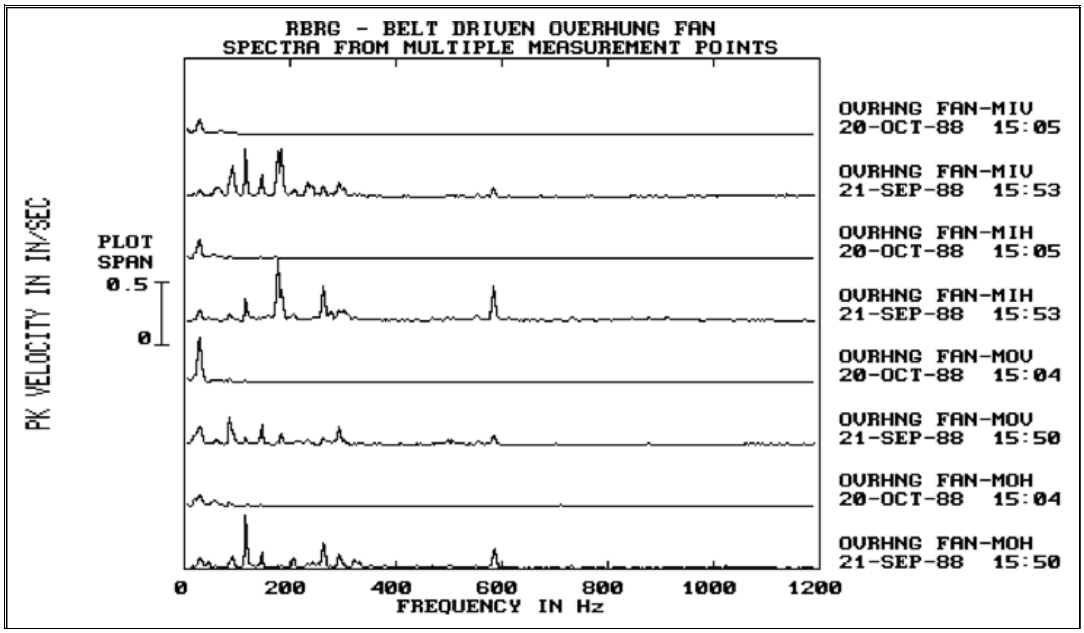


Figure 57

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Vibration in the axial direction of the motor needs further analysis to verify and diagnose the secondary problem.

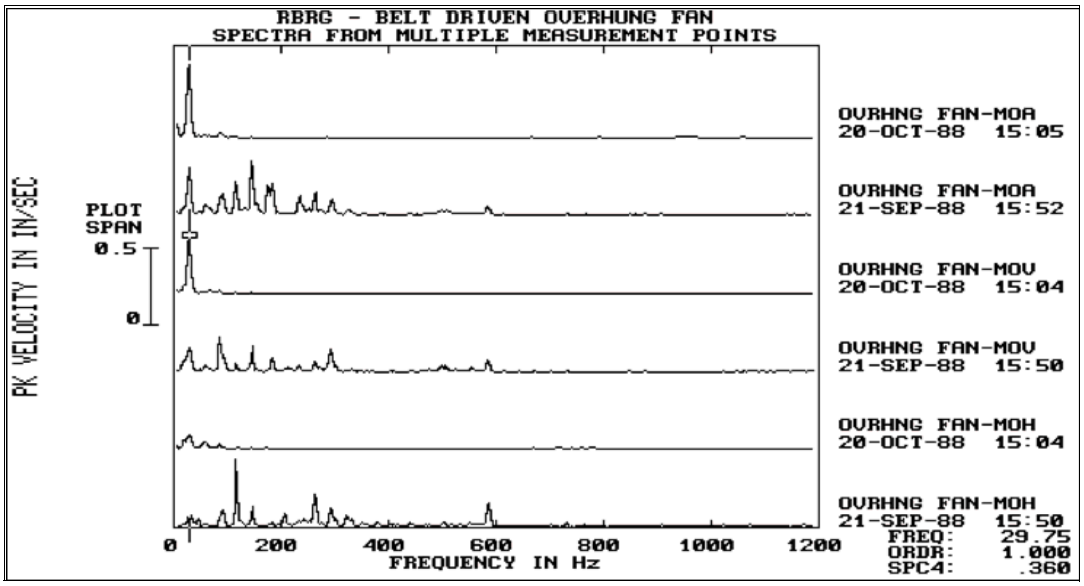


Figure 58

Outboard axial and outboard vertical vibration have both increased on the motor. This combination indicates that an alignment or sheave runout problem now exists on the motor. The motor seems to rock up and down about its inboard feet. The run-out of the motor sheave and the sheave alignment should be checked. The repair, however, seems to have fixed the bearing problem. (Figure 59)

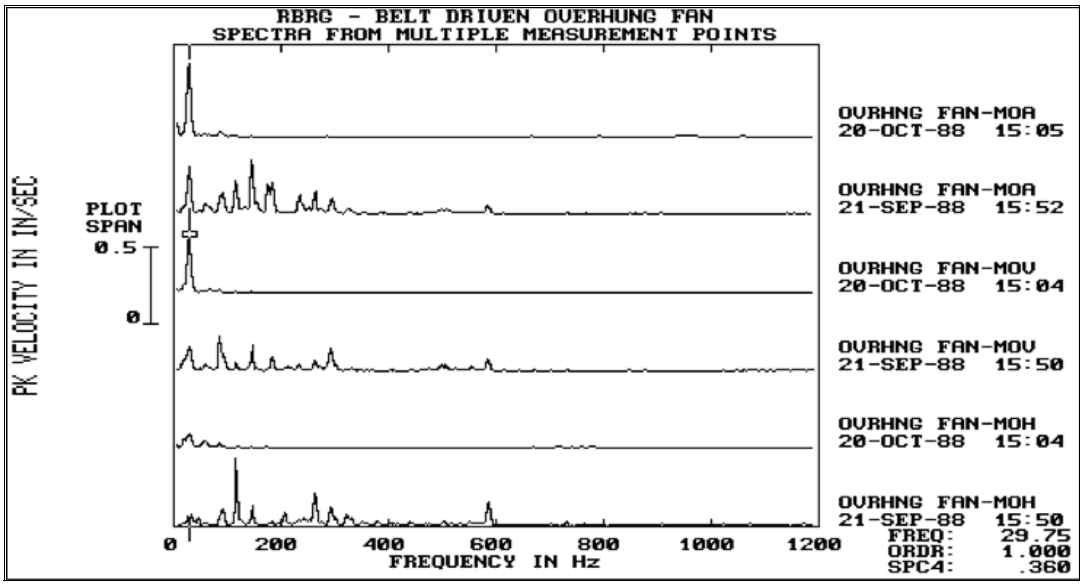


Figure 59

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #12 - Paper Machine Press Roll Bearing

Slow Speed, Very Low Amplitude

The turning speed of this roll is 2.56 Hz or 154 rpm. Note the very low amplitude of the amplitude of the overall value at 0.0396 IPS. Without the benefit of specific bearing frequencies assume that the peaks between 35 and 110 Hz are bearing faults. (Figure 60)

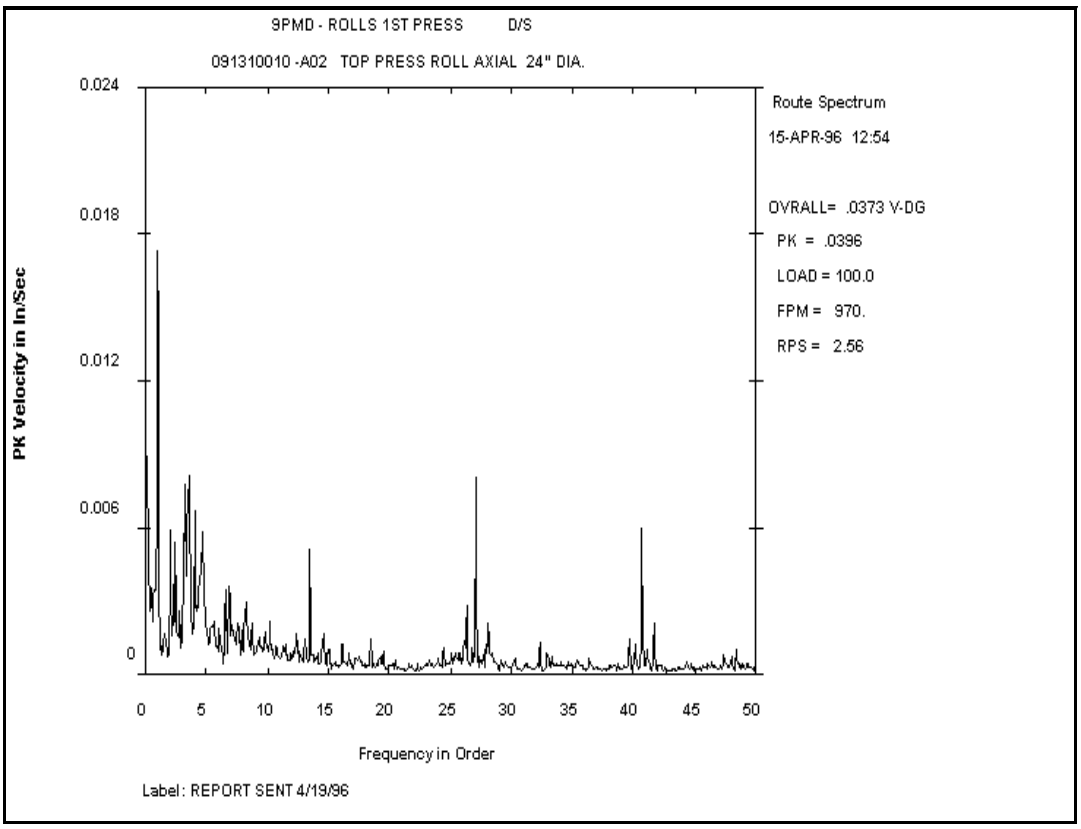


Figure 60

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The advantage of having fault frequencies is highlighted in this case history. Fault Frequencies save the analyst time and takes the some of the guess work out of analyzing vibration problems. Bearing defects and some possible sidebands are evident. (Figure 61)

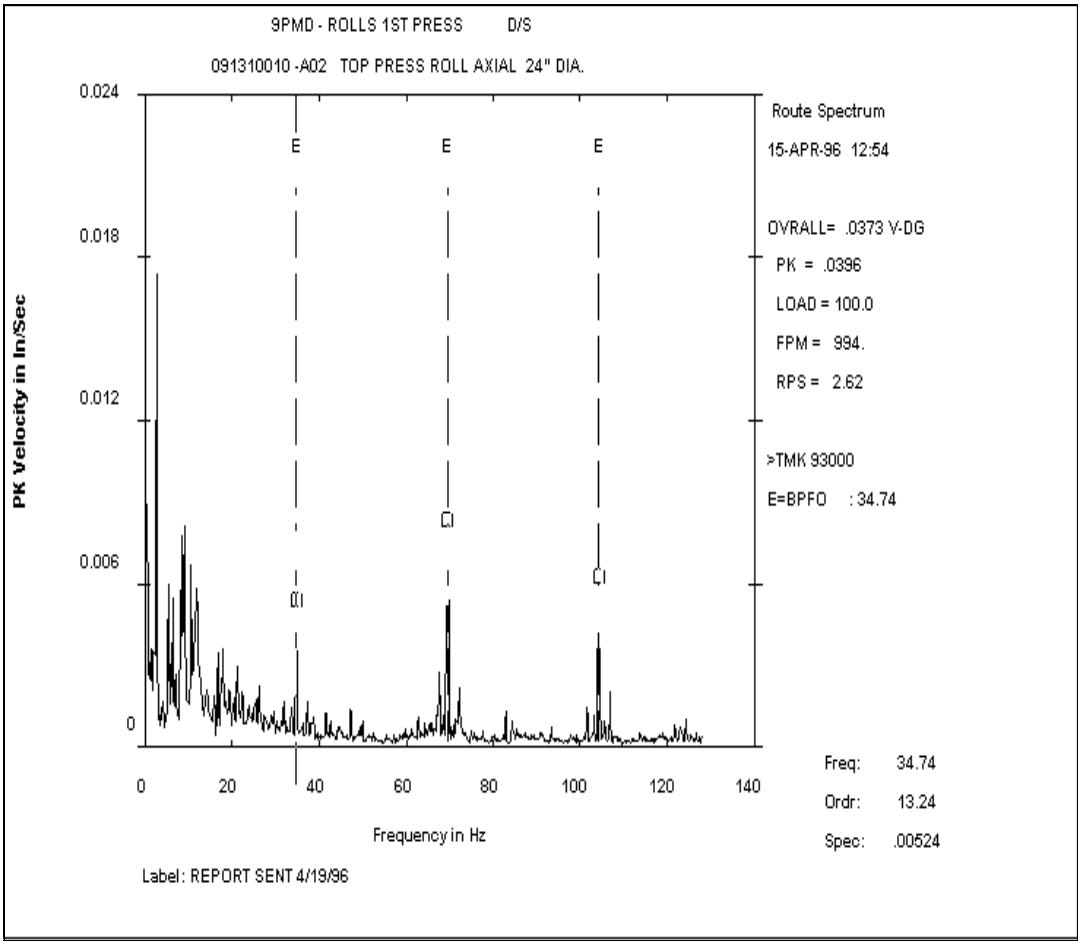


Figure 61

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Even at this slow speed and low G swing in the time waveform in Figure 62, the impacting and the repetitive pattern is typical of a bearing problem. The analyst reported the problem and suggested bearing replacement when the cover repair was scheduled. When the bearing was removed and inspected, a spalled area 1/2" wide by 1 1/2" long was discovered.

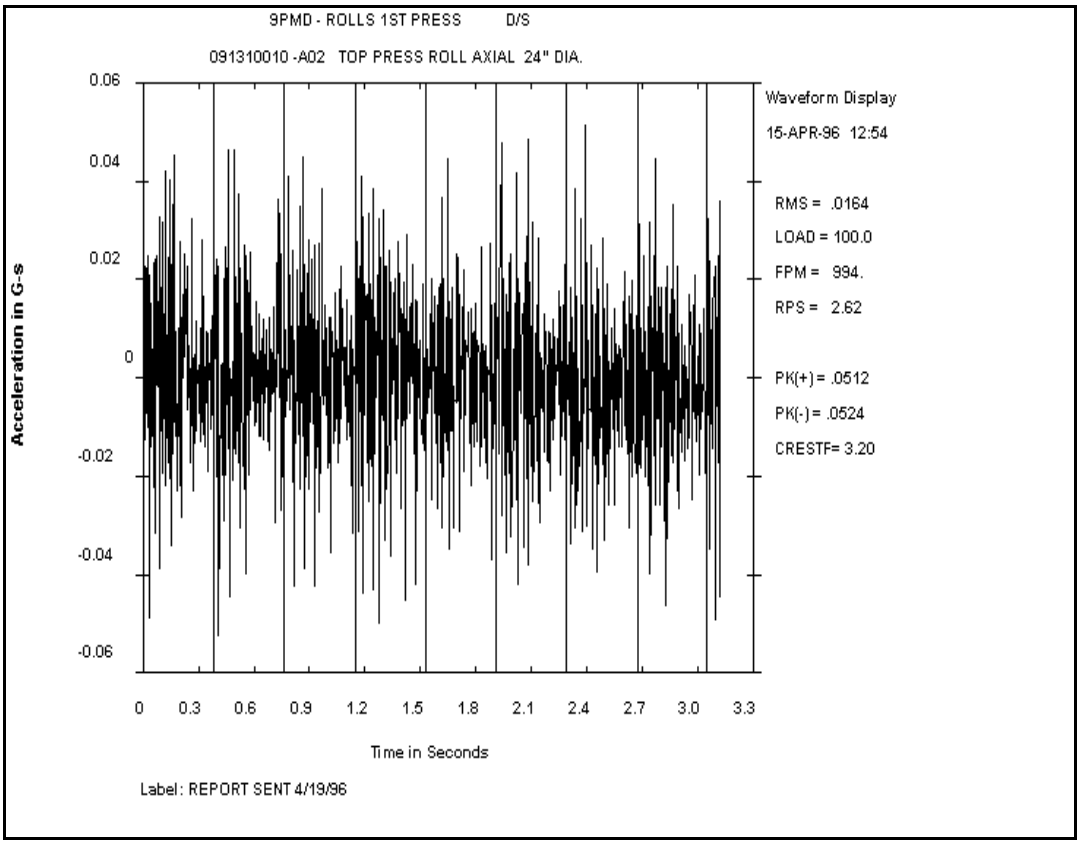


Figure 62

Station:	9 PAPER MACHINE (D/S) WET END
Machine:	091310010 - ROLLS 1ST PRESS D/S
Explanation	** Priority = 4**
VIBRATION IN BEARING ROLL A02	
BEARING HAS OUTER RACE FAULTS WITH HARMONICS RECOMMEND WHEN SCHEDULED TO BE CHANGED TAKE TO MACHINE SHOP AND CHANGE BACK SIDE BEARING	

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #13 - #1 Fire Water Pump

This machine is a direct drive, center hung pump.

Analysis is done without the benefit of fault frequencies.

Turning speed is located at 1749 rpm or 29.15 Hz. A significant peak appears around 15 orders. (See Figure 63)

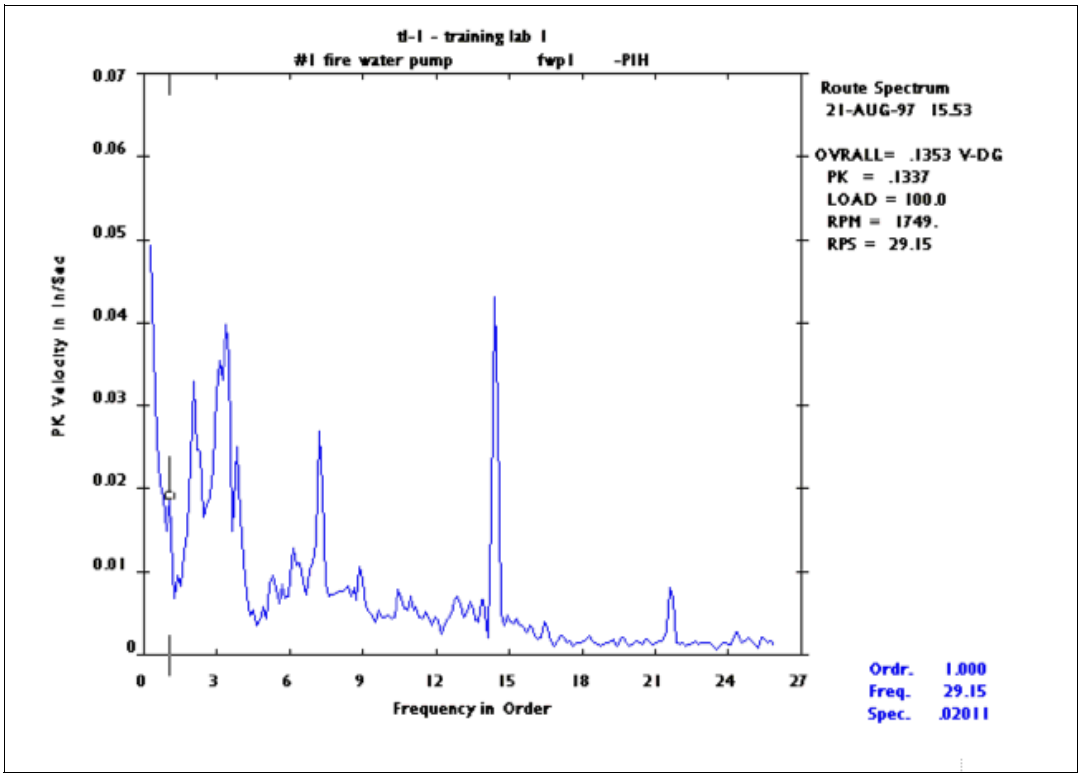


Figure 63

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The spectrum in Figure 64 displays full scale amplitude and frequency as the data was collected. A 50 order F_{max} was selected in the Analysis Parameter Set.

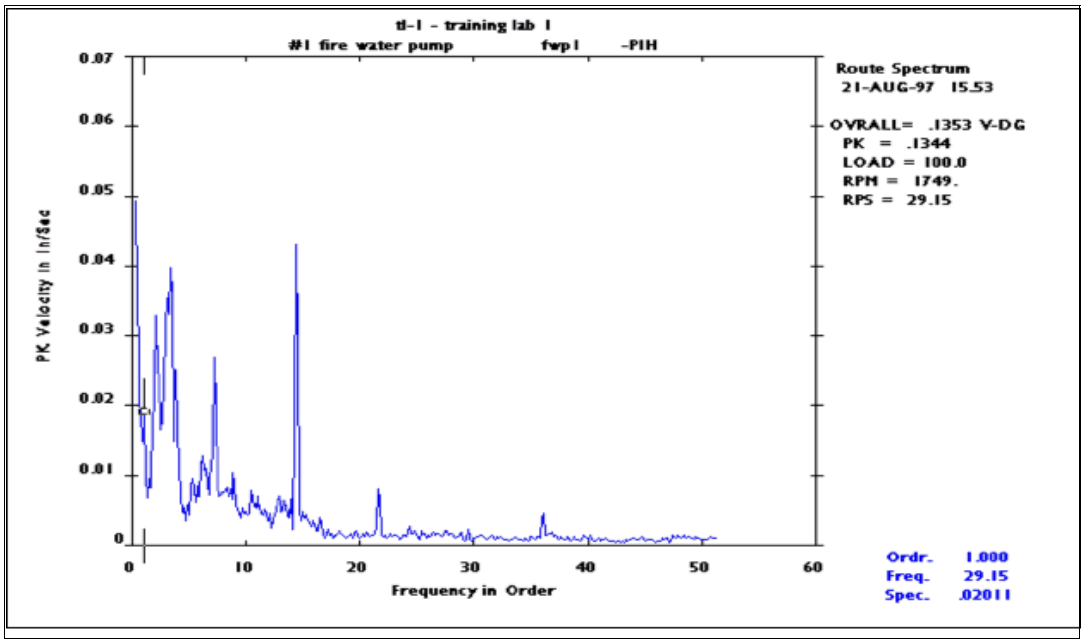


Figure 64

The Peak List in Table 7 displays the distribution of the categories of vibratory energy. The majority is non-synchronous, typical for a rolling element bearing.

LIST OF SPECTRAL PEAKS

Machine: (tl-1) #1 fire water pump
Meas. Point: fwp1 -PIH --> PUMP BEARING INBOARD HORIZONTAL
Date/Time: 21-AUG-97 15:53:28 RPM= 1749. Units=In/Sec PK

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	O V
1	28.85	.0233	.72	13	211.09	.0283	
2	29.11	.0202	1.00	14	226.85	.0085	
3	42.95	.0184	1.47	15	242.16	.0095	
4	61.23	.0346	2.10	16	249.80	.0091	
5	65.76	.0290	2.26	17	268.26	.0116	
6	91.80	.0404	3.15	18	305.24	.0086	1
7	99.19	.0446	3.40	19	317.57	.0074	1
8	111.63	.0261	3.83	20	373.31	.0080	1
9	152.25	.0107	5.22	21	391.46	.0072	1
10	166.30	.0091	5.70	22	406.92	.0078	1
11	178.50	.0143	6.12	23	420.84	.0443	1
12	185.72	.0128	6.37	24	631.35	.0089	2

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.1072	.0233 / 5%	.0202 / 4%	.1027 / 92%
(ELECTRICAL=	0.00E+00/ 0%	FLOOR= .0042	SPEC CREST= 1.6

Table 7

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Harmonics of a defect frequency often indicate a serious or worsening condition. The fundamental cursor has a vertical line that dissects the spectrum. The harmonics appear as "boxes" along the X axis. These in Figure 65 do not seem very significant.

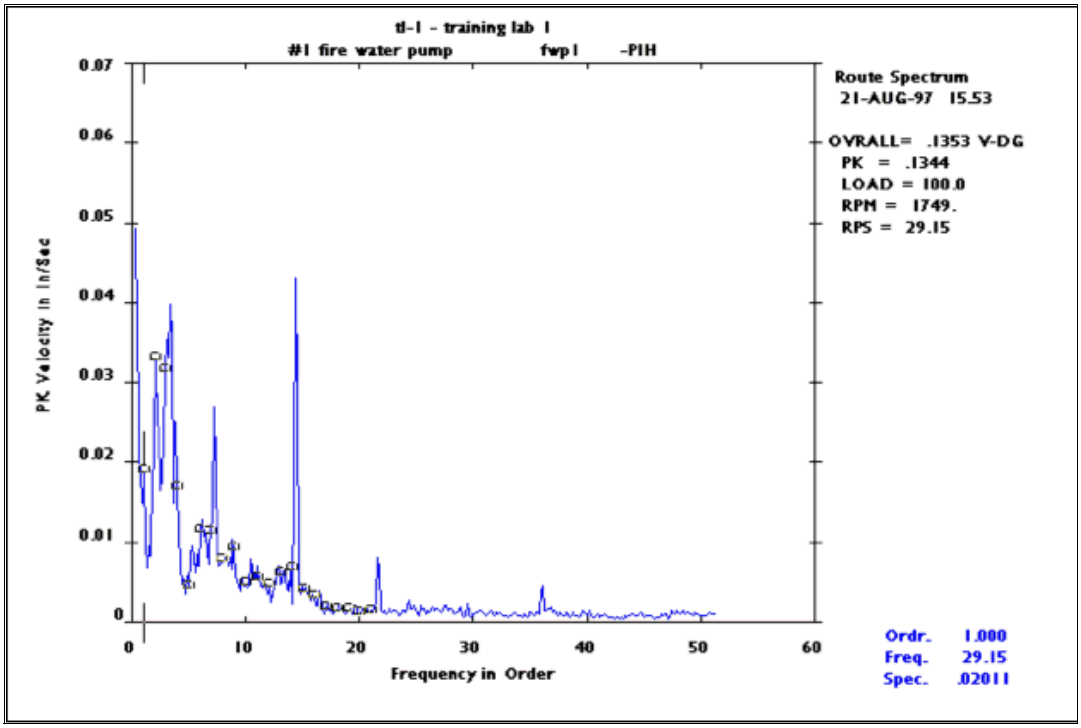


Figure 65

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

After turning speed and those harmonics are located, any other significant peaks should be identified. Harmonics of those frequencies should be found as well. (See Figure 66) The primary cursor is marking a peak at 7.216 orders. Harmonics of this peak are identified with the harmonic cursors. These peaks are nonsynchronous energy. This is a cause for concern since bearing defects generate nonsynchronous energy. There are 5 harmonics of the peak at 7.216 orders. To consider the severity of the defect look at the "overall" reading. The overall is 0.13 in/sec. Consider the speed of the machine, 1749 rpm.

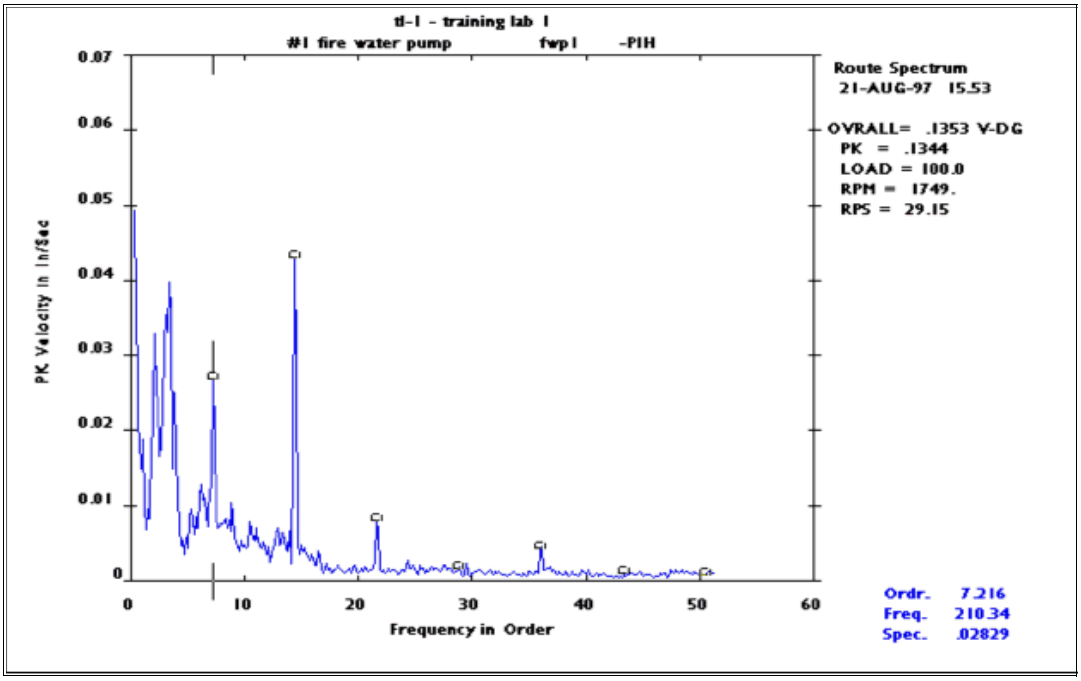


Figure 66

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The time waveform is helpful in identifying a problem and gauging the severity of the defect. The time waveform in Figure 67 reveals at least a 2 "g" swing and a Crest Factor of 3.50.

Armed with a combination of spectral data, Peak-List and time waveform information, the analyst determined there was a bearing problem. The problem was not judged severe enough to warrant a shut down. Sound collection techniques, good trending, and a knowledge of the operating conditions of the machine allowed 30 days lead time to schedule repairs on the machine. It is always prudent to monitor such conditions closely for accelerated bearing degradation.

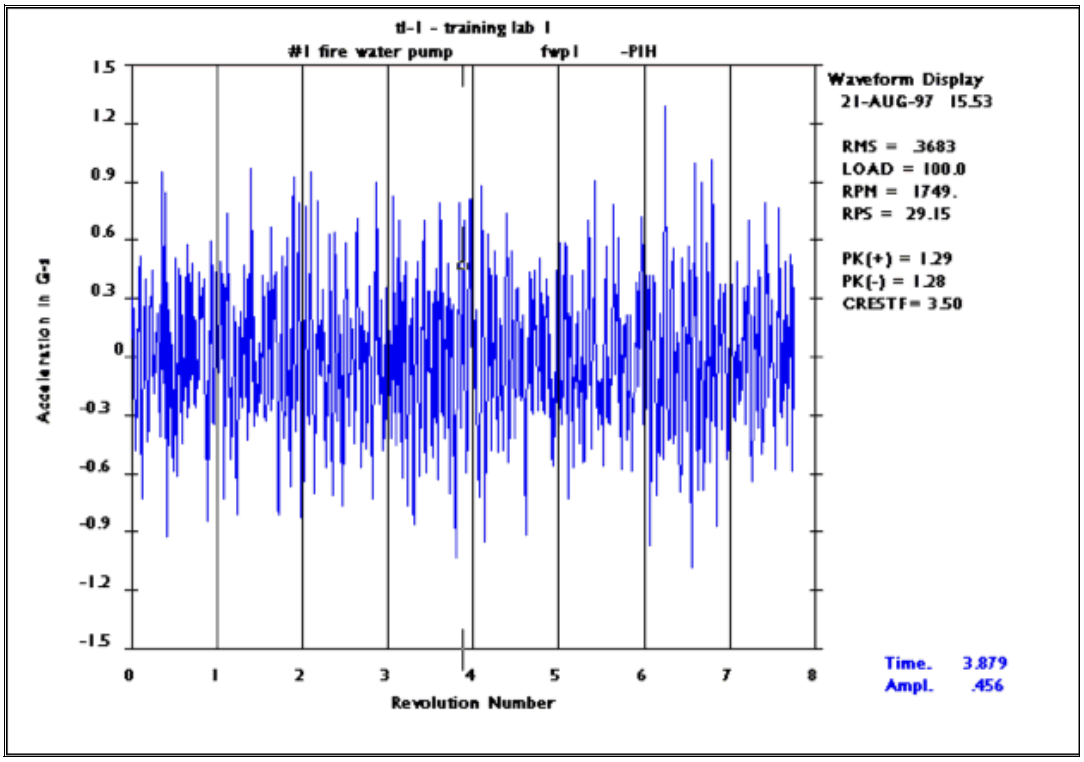


Figure 67

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Case History #14 - Inner Race Defect - #1 Ben Field Pump

- Direct drive
- Center hung

A multiple point spectrum is displayed in Figure 68 to help determine the source of most of the vibration. The motor outboard horizontal position has an elevated turning speed peak, indicating a possible imbalance condition. A single dominant peak of that amplitude can cause other frequencies to be suppressed. A closer examination of the motor points may be in order. Examination of the pump measurement points is necessary as well.

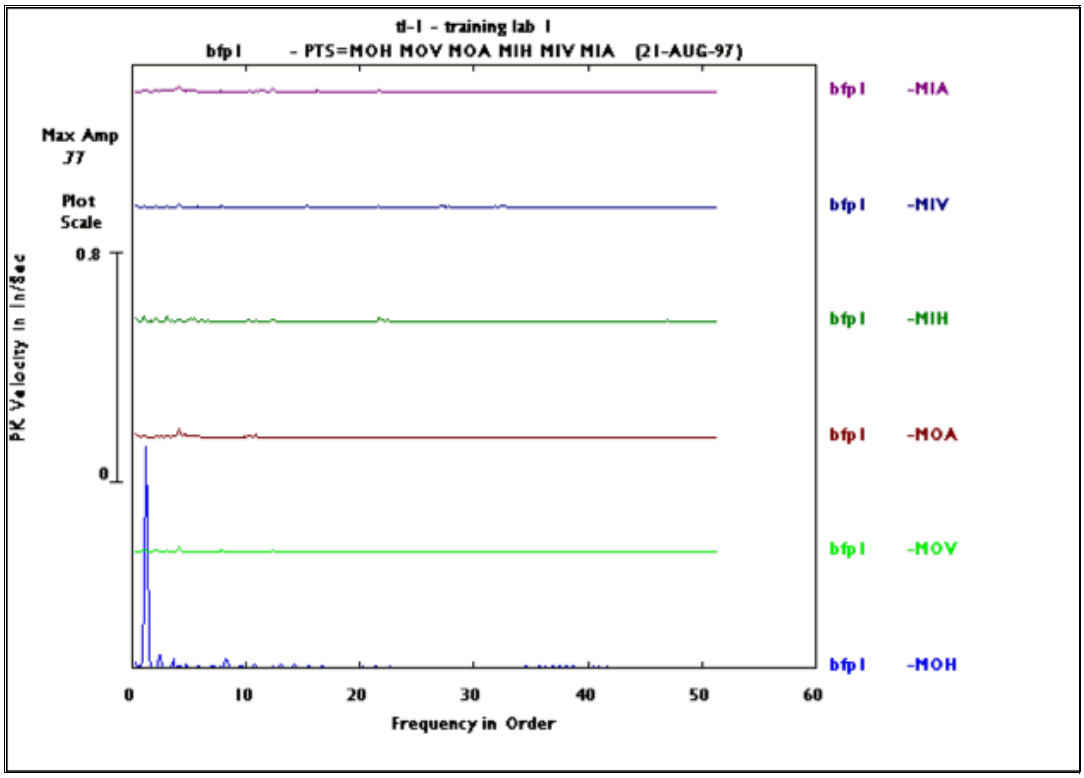


Figure 68

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

The six pump points are displayed in Figure 69 to help determine the source of the high frequency energy. Although the motor outboard horizontal point has higher amplitude at turning speed, more high frequency energy is coming from the pump.

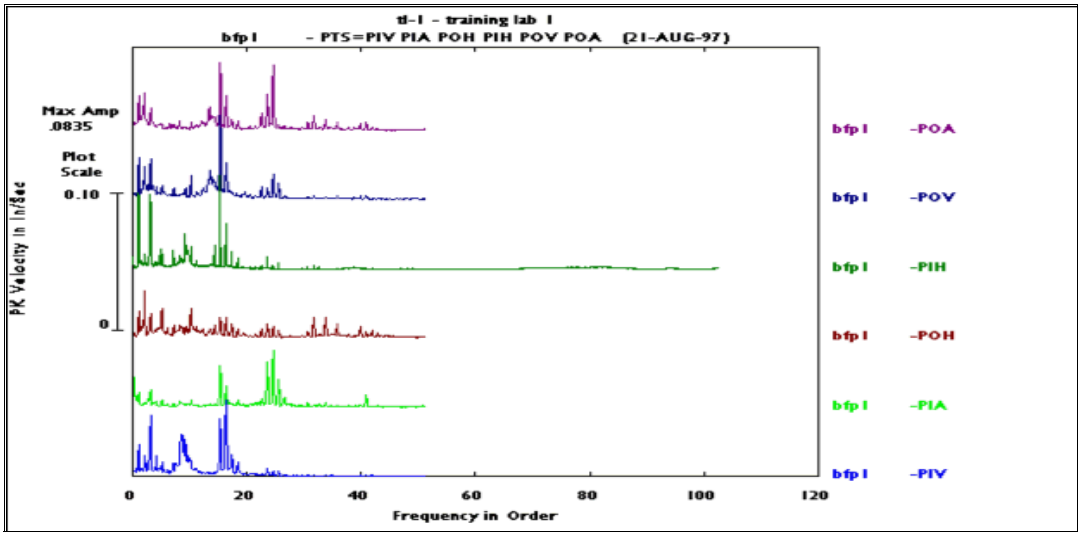


Figure 69

Individual point analysis will help determine which bearing is in worse condition. Examining the pump inboard horizontal point in Figure 69, the evidence shows a significant amount of broad band energy between one and thirty orders. The elevated noise floor indicates a probable looseness condition. Examine the single spectrum in Figure 70 from the pump inboard horizontal measurement point.

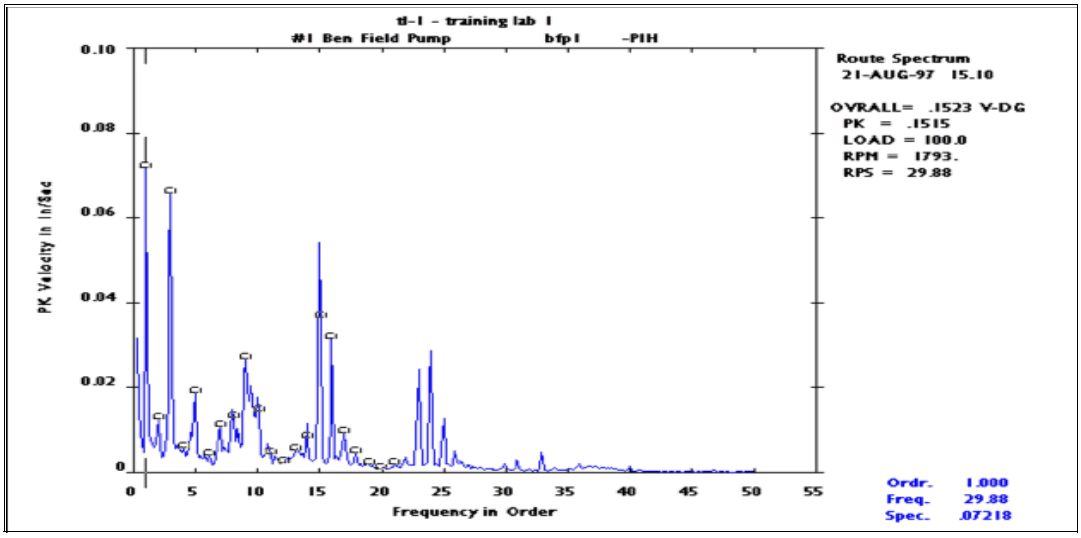


Figure 70

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

In Figure 71 the primary cursor is marking turning speed at 1790 rpm, with harmonics. There are other peaks that require further investigation.

There are also harmonics of a peak at 2.993 orders. Examine the peaks on both sides of these harmonic peaks. If they are evenly spaced about these harmonics there is a good possibility this is an inner race problem.

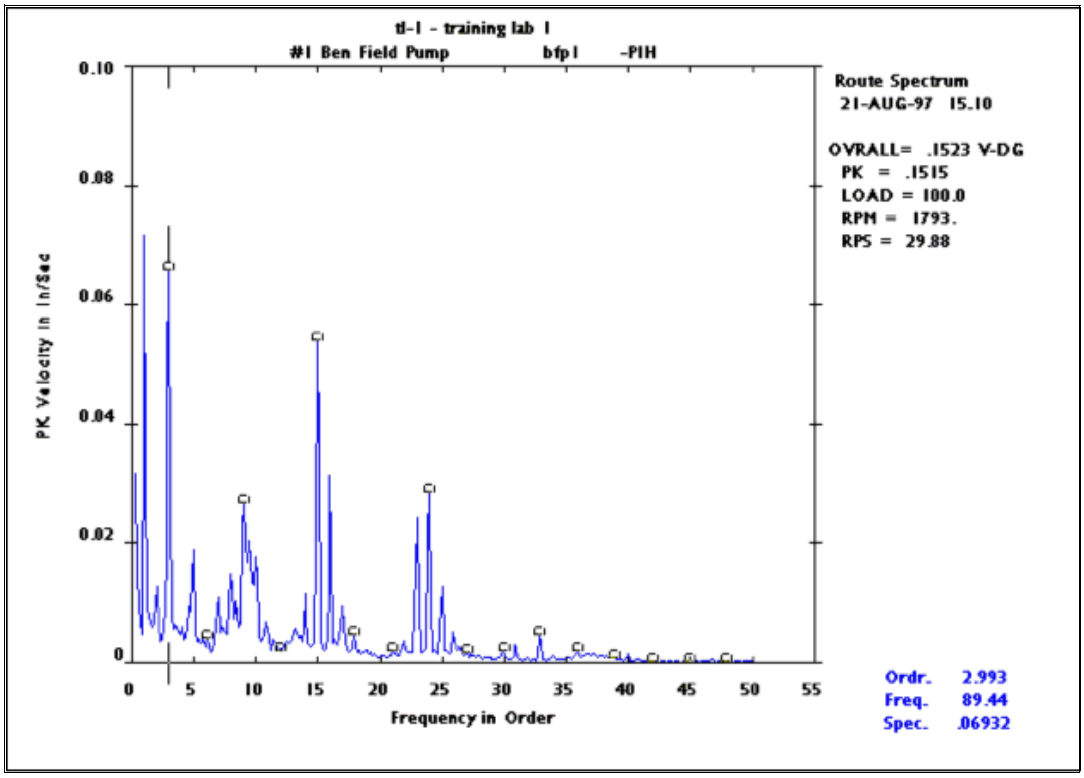


Figure 71

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Examine the time waveform in Figure 72 to help diagnose the problem and determine the severity.

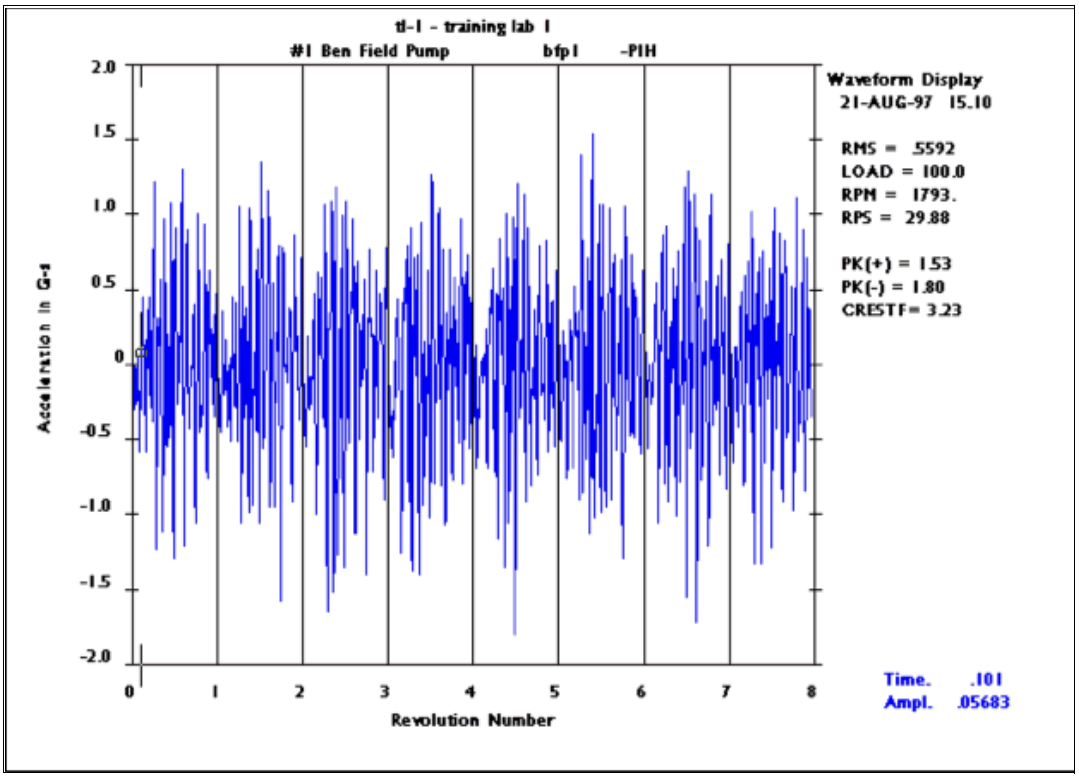


Figure 72

There is some impacting with about a 3 "g" swing with a Crest Factor of 3.23. The analyst determined there was an inner race problem on this pump. Ample lead time was given to prepare for necessary repairs. (See Figure 72)

A successful vibration program adheres to a well developed plan. Practice good, consistent data collection techniques and follow these guidelines:

1. Locate shaft turning speed
2. Look for harmonics of turning speed
3. Identify other significant peaks of interest
4. Locate harmonics of those peaks of interest
5. Check for sidebands
6. Consider the amplitude of the vibration

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

7. Look for elevated noise floor

8. Change the spectral amplitude units

9. Take into account the speed of the machine

10. Factor load as an operational parameter

11. Consider the category of the vibration energy

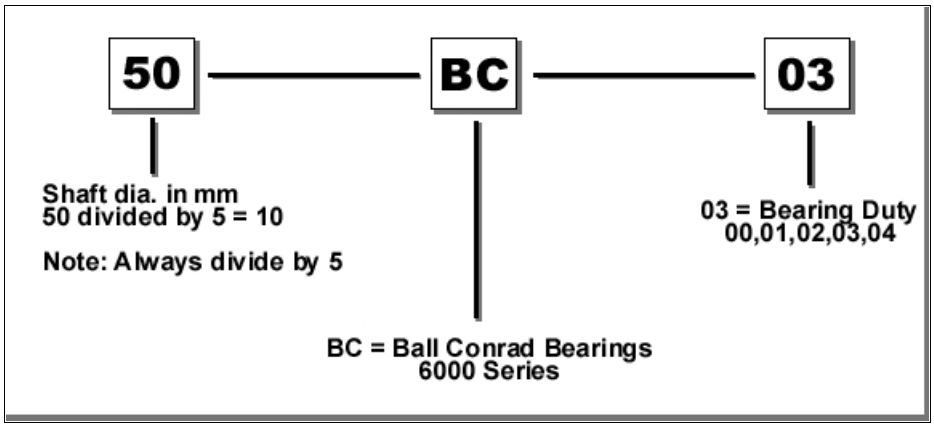
12. View the time waveform in acceleration to show impacting

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

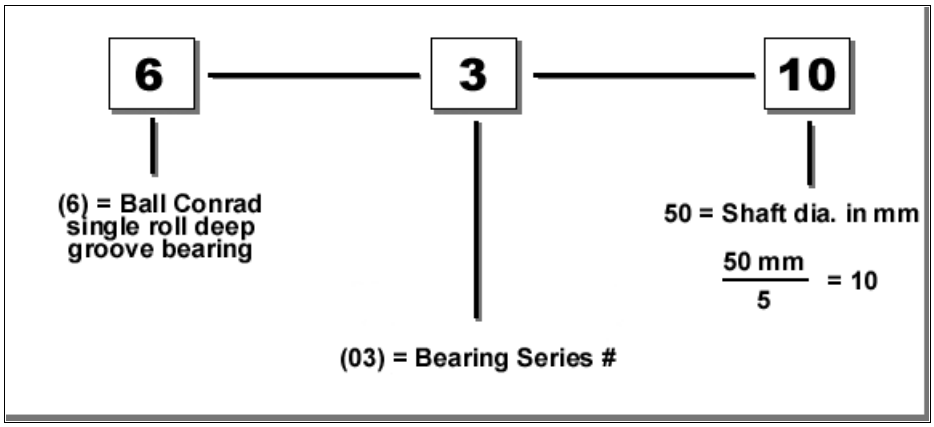
Bearing ID Interpretation

AFBMA Bearing Number = 50 BC 03

50 BC 03 = SKF 6310



AFBMA ID Interpretation



Manufacturer ID Interpretation

Note

Because most motor bearings are listed on motor tags as AFBMA or ABMA numbers, you may be able to calculate the common bearing number given the AFBMA on the equipment nameplate.

Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Bearing Interchange

Address:

AFBMA Conversion
Interchange Inc.
P.O. Box 16244-B
St. Louis Park, MN 55416
Phone - (612) 929-6669
Phone - (800) 669-6208 (Toll Free)
Fax - (612) 929-0395
Fax - (800) 729-0395 (Toll Free)

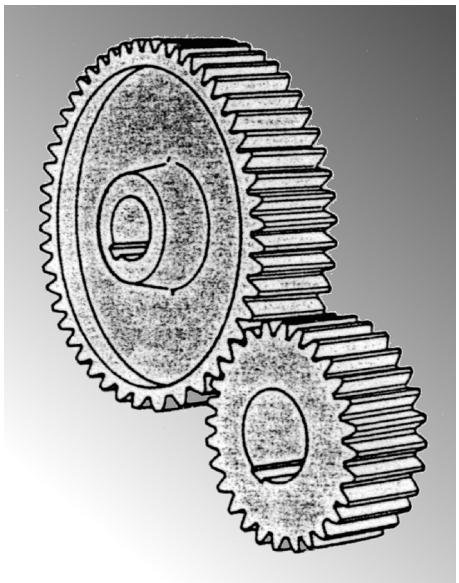
Copyright 2006, Emerson Process Management, All rights reserved. Rev 04/06

Gear Defects

Section 6

Objectives

- Define and Calculate gear mesh frequency.
- Determine Some Causes of Gear Box Defects
- Identify the Characteristics of Gear Mesh Problems
- Establish some Corrective Actions.



Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Gear mesh

Gear mesh may be defined as the action of two gears rotating so that they share a common tangent during rotation. Figure 1 illustrates two gears meshing together. Some typical applications of gear boxes are reverse in shaft rotation, speed increase or decrease, a change in angle, increase in torque, etc. Depending on the type of gears in the sets, location of measurement points and loading parameters, data collection on some gear boxes can be challenging.

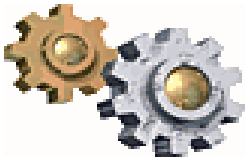


Figure 1

Gear Box Defects

Several problems may arise in gear boxes. Misaligned gears, chipped or broken teeth, stress fractures, and worn gears are all common to gear drives. Among the causes of premature gear box failure are:

- Improper lubrication
- Wrong application
- Bearing failure
- Water intrusion
- Overheating
- Poor craftsmanship

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Characteristics of Gear Mesh Frequency Issues

Spectrum:

- High frequency / low amplitude synchronous peaks
- Harmonics of GMF with turning speed sidebands
- Sideband amplitude(s) increase as the condition deteriorates

Waveform:

- Impacting* (normal for gearboxes)
- Very busy waveform
- Once per revolution pulses with cracked or broken tooth

*Since gearboxes are naturally energetic due to the gears meshing, trending is critical.

Gear Mesh Frequency Calculation

GMF =	Gear Mesh Frequency in Hz
RF =	Rotational Speed of Gear in Hz
GT =	# of Gear Teeth

$$GMF = RF \times GT$$

For example:
A gear is rotating at 10 Hz with 72 teeth:

$$GMF = 10Hz \times 72 = 720Hz$$

Caution!
Use the proper units when calculating gear mesh! Gear mesh is never expressed as RPM.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Gear Ratio Calculation

Often, in a simple, single reduction gear box, knowing the ratio of the input to output gears can be very helpful in determining the turning speed of the output shaft.

$$Ratio = \frac{Number\ of\ teeth\ on\ gear\ on\ input\ shaft}{Number\ of\ teeth\ on\ gear\ on\ output\ shaft}$$

For example, in a gear box with 72 teeth on the input gear, and 24 teeth on the output gear the ratio would be 3:1.

$$\frac{72\ teeth\ on\ input\ gear}{24\ teeth\ on\ output\ gear} = 3$$

The output shaft would rotate 3 times faster than the input shaft.

In the case of having 24 teeth on the input shaft and 72 on the output shaft, the output speed would be 1/3 that of the input, a 1:3 ratio.

$$\frac{24 teeth\ on\ input\ gear}{72 teeth\ on\ ouput\ gear} = \frac{1}{3}$$

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

To calculate the speed using the ratio only, multiply the input speed by the ratio of the gear box.

For example: If the ration is 3:1, and in input speed of 10 Hz, the output speed would be 30 Hz.

$$OutputSpeed = \frac{3}{1} \times 10Hz = 30Hz$$

If the input shaft has 24 teeth rotating at 30 Hz and the output shaft has 72 teeth, the output speed would be 10 Hz.

$$\frac{24 teeth\ on\ input\ shaft}{72 teeth\ on\ ouput\ shaft} \times 30Hz = 10Hz$$

Corrective Actions

On most gear boxes, it is best to replace entire gear sets instead of just the faulty gear. After running for many hours, gear trains tend to generate a wear pattern. When a new gear is introduced into a gear box with extensive operating time on it, new problems may arise. In addition to replacing the gears, bearing replacement is also suggested.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Calculating Gear Box Output Speed

Method #1

Single Reduction Example:

◆ 25 teeth on the gear on the input shaft
◆ Input shaft speed = 1750 cpm or 29.2 Hz
◆ 17 teeth on the gear on the output shaft

Objectives:

- **Determine GMF**
- **Calculate output shaft speed**

$$GMF = Number\ of\ teeth\ on\ input\ shaft \times turn\ speed$$

$$Output\ speed = \frac{GMF}{Number\ of\ teeth\ on\ output\ shaft\ gear}$$

$$GMF = 25\ teeth \times 1750\ cpm\ OR\ 29.2\ Hz = 43,750\ cpm\ OR\ 729.2\ Hz$$

$$GMF = 43750\ cpm\ OR\ 729.2\ Hz$$

$$Output\ Speed = \frac{43750\ cpm\ OR\ 729.2\ Hz}{17\ teeth} = 2573.5\ cpm\ OR\ 42.9\ Hz$$

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Method #2

Determine Gear Mesh Frequency of this gearbox

Single Reduction Example:

◆ 25 teeth on the gear on the input shaft
◆ Input shaft speed = 1750 cpm or 29.2 Hz
◆ 17 teeth on the gear on the output shaft

$$GMF = Number\ of\ teeth \times turn\ speed$$

$$GMF = 25T \times 1750\ cpm\ OR\ 29\ Hz = 43750\ cpm\ OR\ 729.2\ Hz$$

Determine the speed of the output shaft

To calculate the speed of the output shaft when the following are known:

- The input shaft speed and the number of teeth on the input gear
- The number of teeth on the gear on the output shaft

Calculate the Gear Ratio

$$Ratio = \frac{Number\ of\ teeth\ on\ the\ input\ shaft\ gear}{Number\ of\ teeth\ on\ the\ output\ shaft\ gear}$$

$$Ratio = \frac{25\ teeth}{17\ teeth} = 1.47$$

$$Ouput\ speed = 1.47 \times 1750\ cpm\ OR\ 29.2\ Hz$$

$$Output\ Speed = 2572.5\ cpm\ OR\ 42.9\ Hz$$

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Gears

- Gear mesh energy appears regardless of gear condition.
- Problems are indicated by gear mesh harmonics with sidebands.
- The sidebands around gear mesh frequency will be spaced equal to the turning speed of the shaft that the bad gear is on. The sidebands are modulated by the turning speed of that shaft.
- Gear mesh is synchronous energy.
- An increase in amplitudes of gear mesh sidebands indicates problems.
- To calculate gear mesh frequency

GMF = the number of teeth x turn speed of the gear (shaft)

- Gear misalignment appears as 2 x GMF in the spectrum.
- A recommended F_{max} to analyze gear problems
  - 3.5 x GMF if possible, the extra 0.5 allows sidebands in the data
  - Accelerometer type and mounting method is critical
- General guidelines for choosing Lines of Resolution
  - 1600 lines of resolution if the F_{max} is less than 2,000 Hz.
  - 3200 lines of resolution if the F_{max} is greater than 2,000 Hz.
- Gear replacement suggestions
  - Mark gears - so they can be reinstalled in same mesh, if reusing some gears.
  - It is best to replace gears in sets.
- Other technologies for good gearbox monitoring.
  - Oil Analysis
  - Infrared
  - Ultrasonics

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The spectrum in Figure 2 is from a machine with a gear problem. The sidebands are equally spaced at the rotational frequency of the shaft carrying the defective gear. In this example, the pinion was the bad gear.

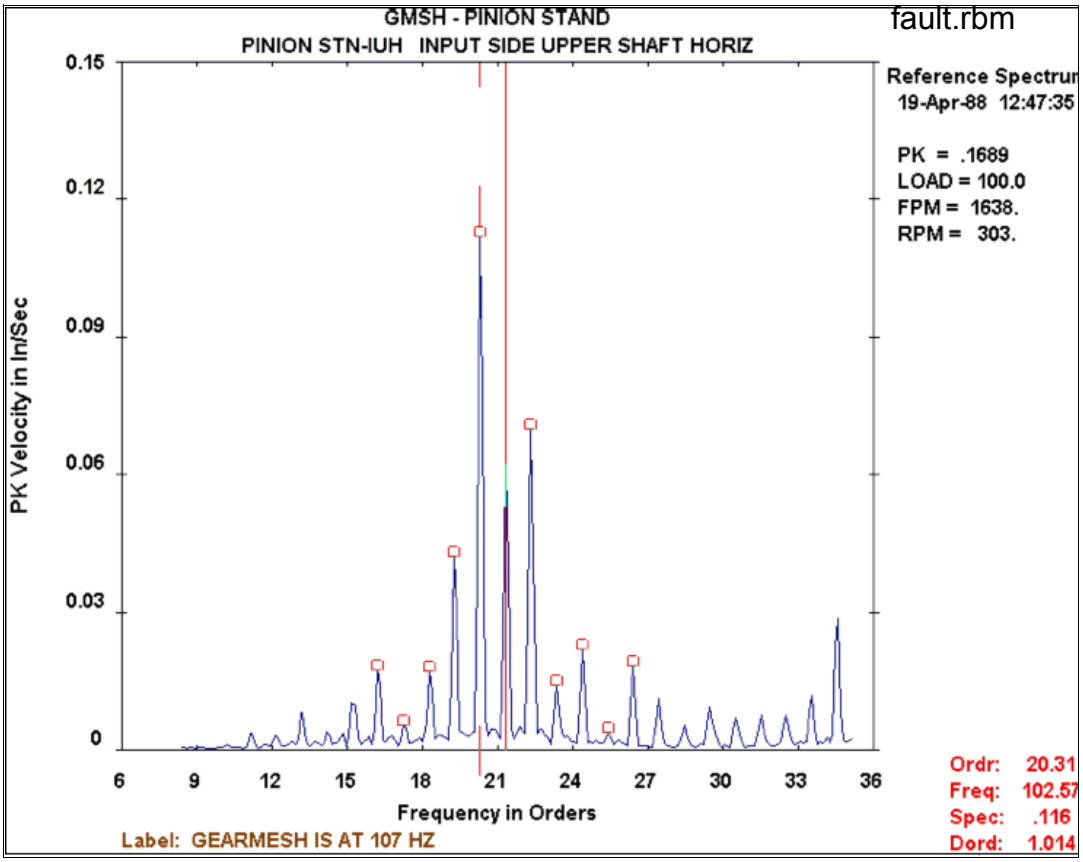


Figure 2

The sideband spacing is listed as the Dord: (differential order) at the lower right hand corner of the Spectrum.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The time waveform pattern in Figure 3 shows the impacting of the gears when they come into mesh. This time waveform is in velocity. Acceleration waveforms indicate the presence of impacting better than velocity. Therefore gearbox waveforms should be viewed in acceleration.

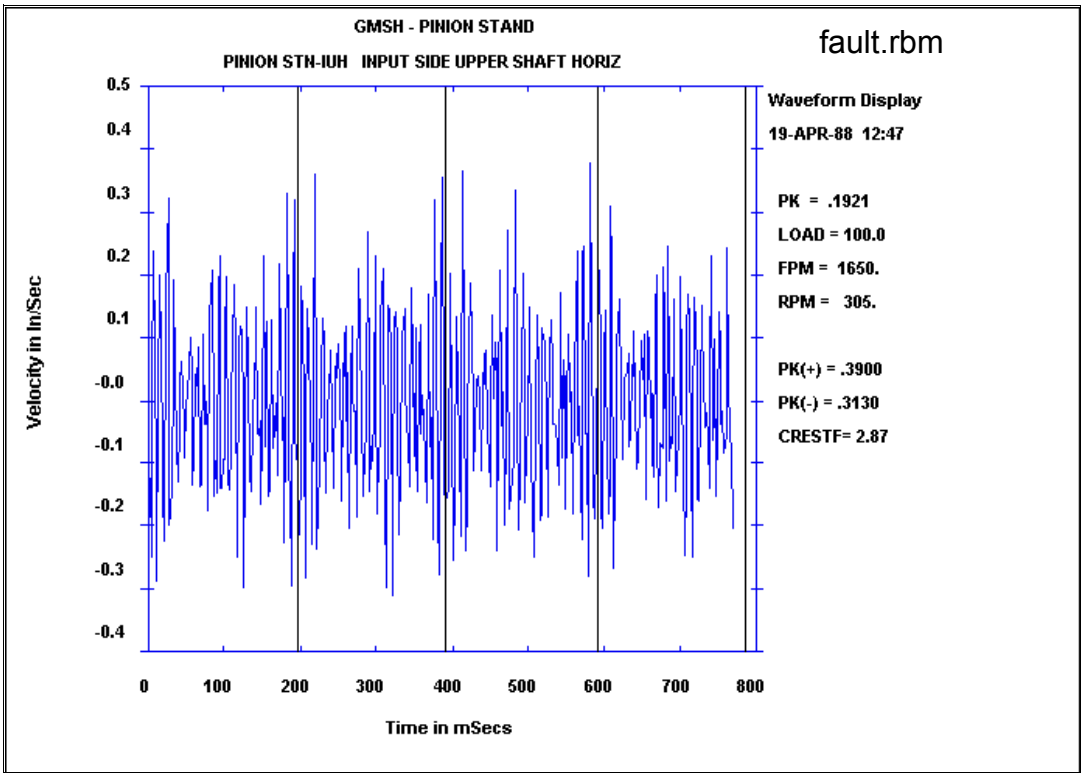


Figure 3

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Objective:

Determine the output speed of the shafts in the examples in Figures 4 and 5.

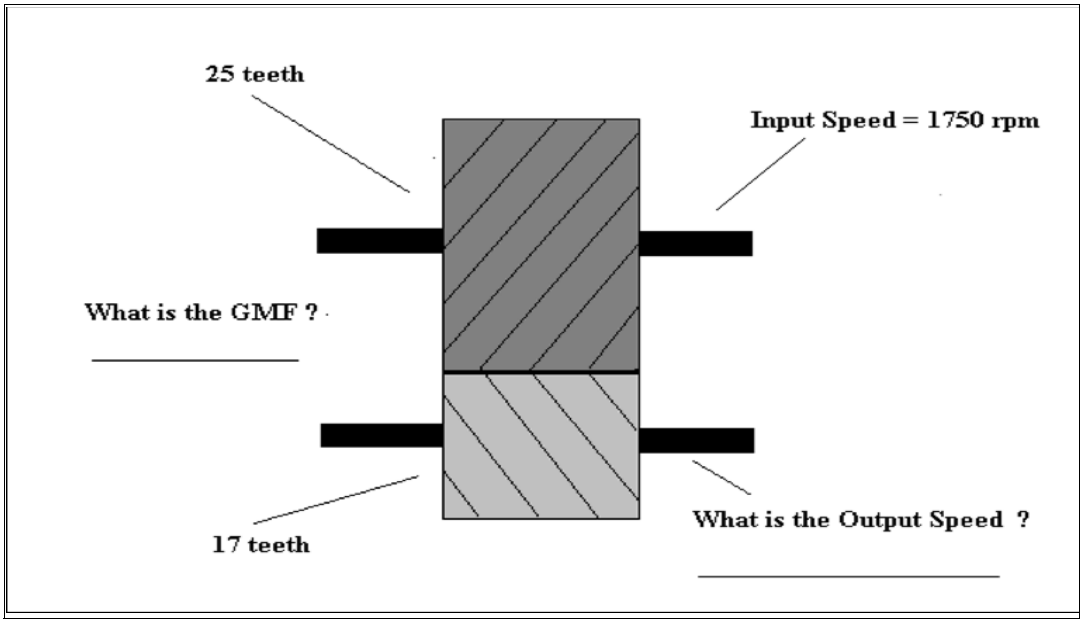


Figure 4

Notice the intermediate shaft in Figure 5. Treat it as an output from the original gear. Then treat it as the input for the final gear shaft.

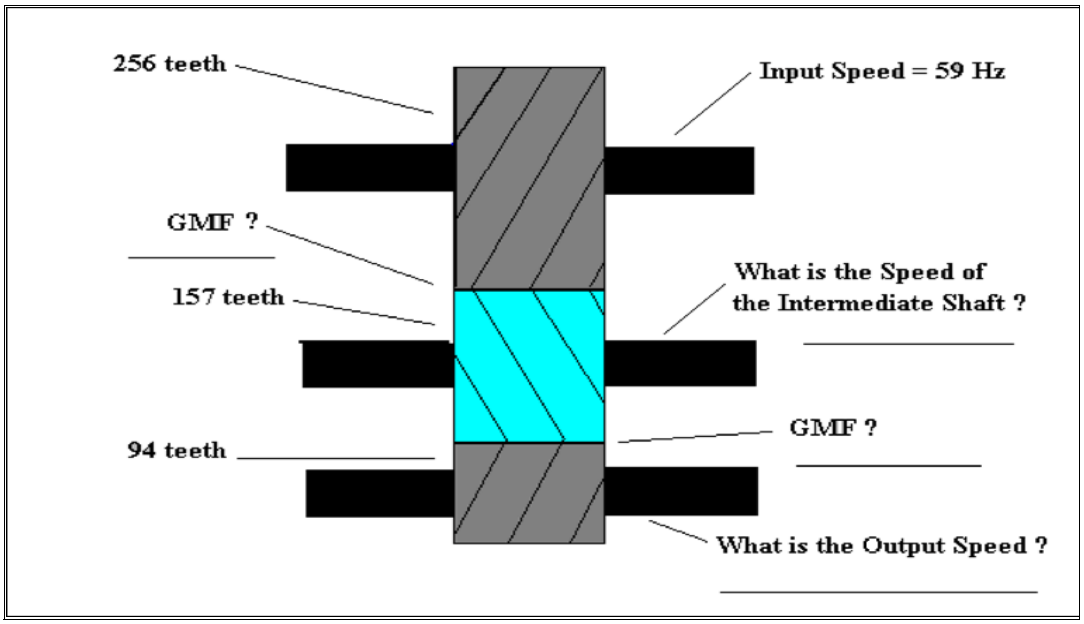


Figure 5

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Objective:

Calculate the output shaft speed in the exercises in Figures 6 and 7. Following the same convention as in the examples in Figures 4 and 5 help keep the train in proper order.

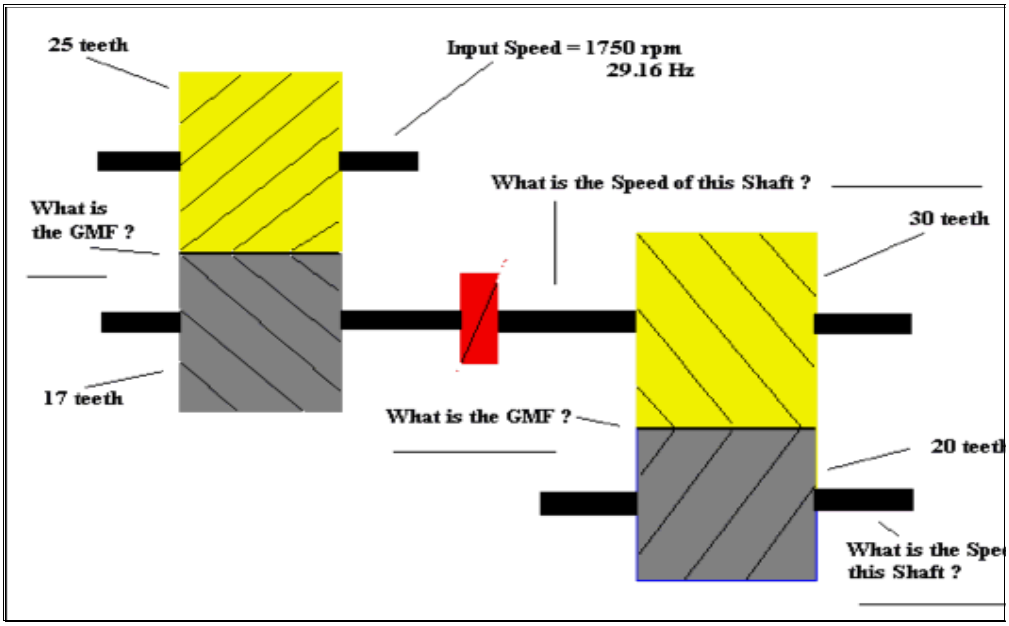


Figure 6

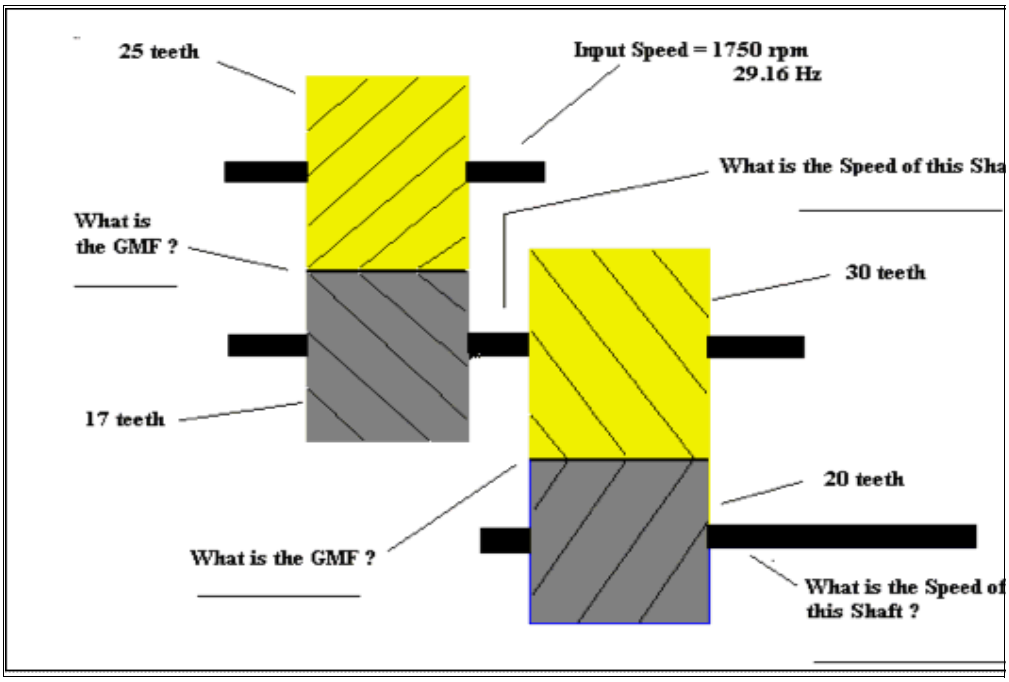


Figure 7

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Gear Signatures

The complex mechanical nature of gearboxes makes the interpretation of gear signatures difficult.

A few concepts help recognize gear problems.

Gear Mesh Frequency (GMF)

- Appears regardless of gear condition
- Amplitude changes significantly with load

Sidebands

- High amplitude sidebands indicate a problem(s)
- Sidebands indicate which gear is bad by the spacing between peaks

Gear natural frequency

- Gear natural resonance excited by gear defects
- Good indicator of a problem

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Gear Mesh

Gear defects produce low amplitude vibration at high frequencies similar to rolling element bearing defects. They differ in that gear problems are synchronous and rolling element bearings are nonsynchronous. Gear defects are manifest at GMF and/ or harmonics of GMF. Calculate GMF by multiplying the number of teeth on a given gear times its turning speed.

$$GMF = \text{Number of teeth on gear} \times \text{Turning speed of gear}$$

For example, a 256-tooth gear rotates at 3600 RPM or 60 Hz.

$$GMF\ (Hz) = 256 \times 60 Hz = 15360 Hz$$

$$GMF\ (CPM) = 256 \times 3600\ RPM = 921600\ CPM$$

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Sample Problem

Calculating GMF in Hz helps keep the values smaller and easier to work with. When analyzing a multiple gear train, additional parameters need to be included to identify which gears show defects. A sample problem and gear train are given in Table 1. Gear frequency calculations, including sidebands, are included.

Drive gear (input) speed	**=**	**60 Hz**
# of teeth on drive gear	**=**	**256**
# of teeth on first pinion	**=**	**38**
# of teeth on second pinion	**=**	**21**
GMF	**=**	**15,360 Hz**

Table 1

$$First\ pinion\ shaft\ speed = \frac{GMF}{Number\ of\ teeth\ on\ first\ pinion}$$

$$First\ pinion\ shaft\ speed = \frac{15360}{38}$$

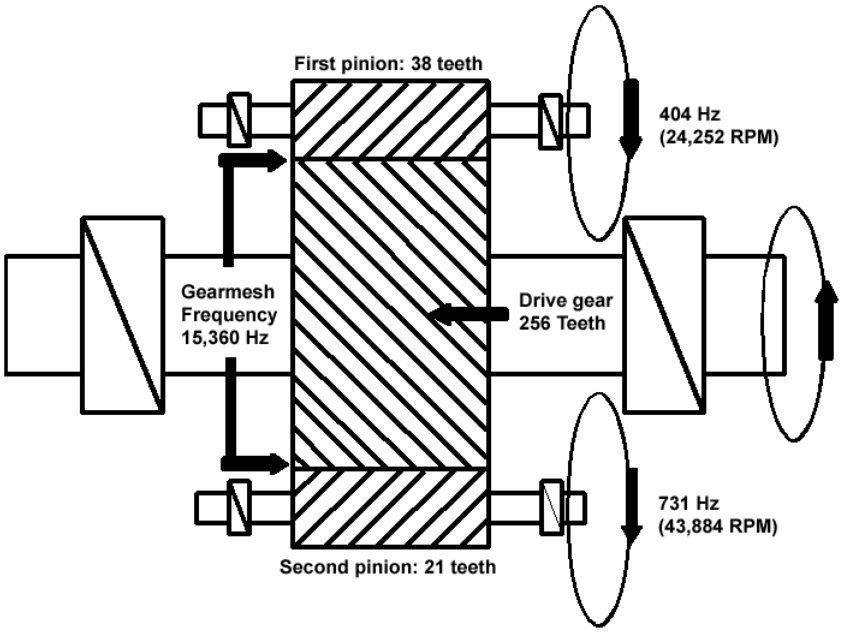
$$First\ pinion\ shaft\ speed = 404.2Hz = 24252\ RPM$$

$$Second\ pinion\ shaft\ speed = \frac{GMF}{Number\ of\ teeth\ on\ second\ pinion}$$

$$Second\ pinion\ shaft\ speed = \frac{15360}{21}$$

$$Second\ pinion\ shaft\ speed = 731.4 = 43884\ RPM$$

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06



Teeth (T1) on Gear 1:	256
Teeth (T2) on Gear 2:	38
Gear Ratio (T2/T1)	.1484
Largest Common Factor	2

Harmonics	Shaft 1	Shaft 2	Gear Mesh	Ass. Phase	Tooth Rept
1	60.00	404.21	15360.00	7680.00	3.16
2	120.00	808.42	30720.00	15360.00	6.32
3	180.00	1212.63	46080.00	23040.00	9.47
4	240.00	1616.84	61440.00	30720.00	12.63

Harmonics	Gear mesh - Plus Shaft 1 Sidebands		Gear mesh- Plus Shaft 2 Sidebands	
1	15360.00		15360.00	
	15300.00	15420.00	14955.79	15764.21
	15240.00	15480.00	14551.58	16168.42
2	30720.00		30720.00	
	30660.00	30780.00	30315.79	31124.21
	30660.00	30780.00	29911.58	31528.42
3	46080.00		46080.00	
	46020.00	46140.00	45675.79	46484.21
	45960.00	46200.00	45271.58	46888.43
4	61440.00		61440.00	
	61380.00	61500.00	61035.79	61844.21
	61320.00	61560.00	60631.58	62248.43

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Sample Gearmesh Calculation

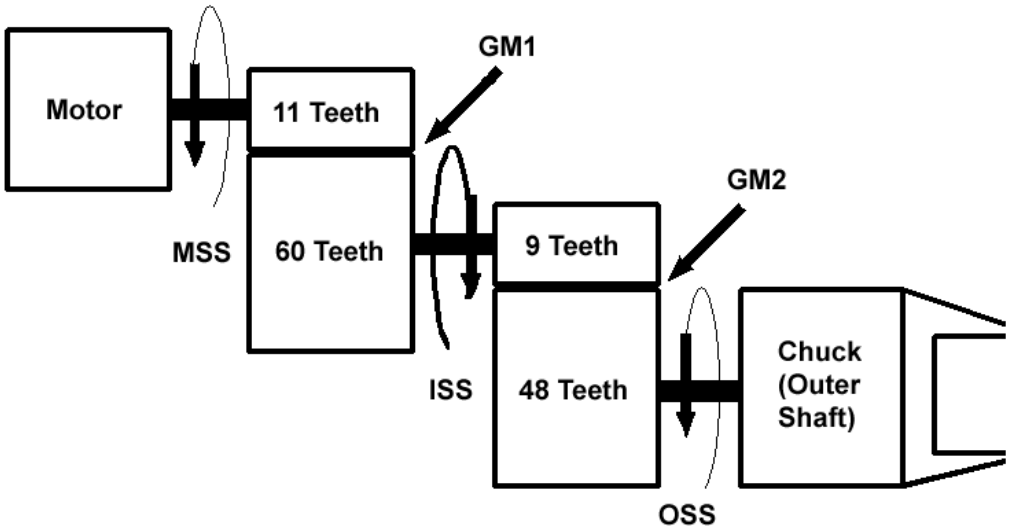


Figure 8

Use the data in Figure 8 to practice calculating complicated gear train frequencies. In this case, output speed is given and the objective is to determine the input speed of the motor.

Output Shaft Speed (measured with tachometer)	=	3.8 Hz = OSS
Gearmesh 2	=	_____ Hz X 48
	=	_____ Hz = GM_2
Intermediate Shaft Speed	=	GM2 / 9
	=	_____ Hz / 9
	=	_____ Hz = ISS
Gearmesh 1	=	_____ Hz X 60
	=	_____ Hz = GM_1
Motor Shaft Speed	=	GM_1 / 11
	=	_____ Hz / 11
	=	_____ Hz = MSS

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Spectrum for Gearmesh - Calculation Example

Relative frequencies can be calculated and predicted on the spectrum by using the sample gear mesh calculation setup from Figure 8. Using 3.8 Hz for the chuck shaft speed the calculated frequencies should approximate those shown by the fault frequencies on the right hand side on the spectrum.

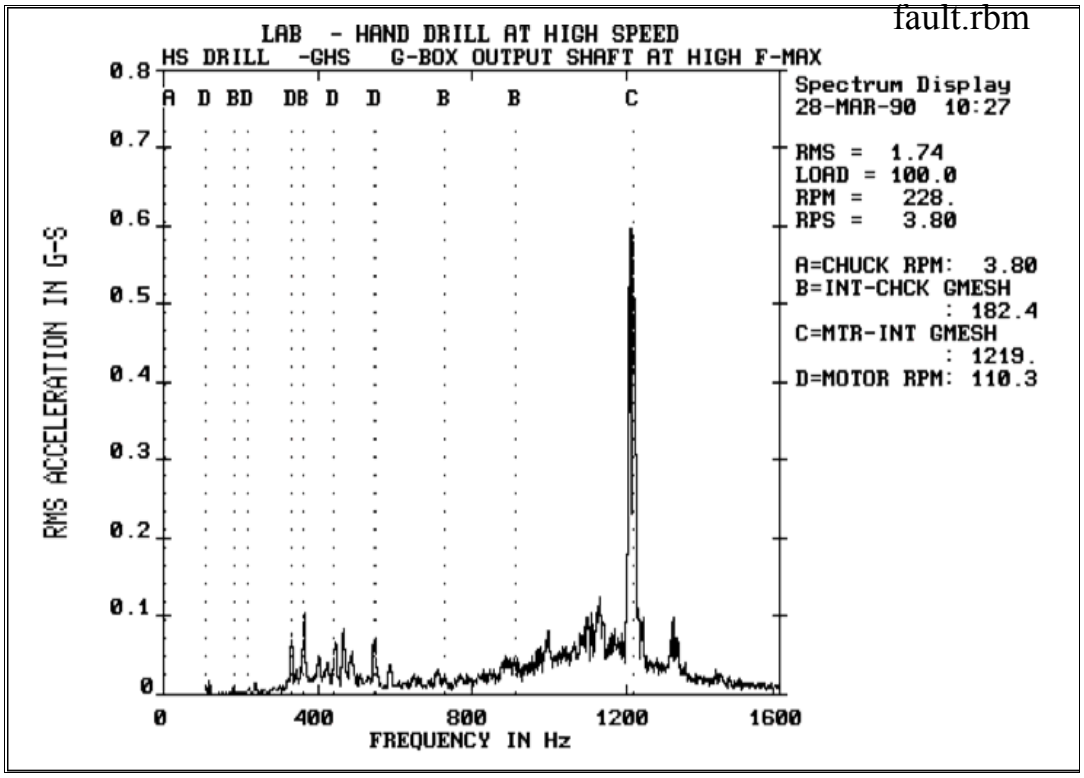


Figure 9

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #1

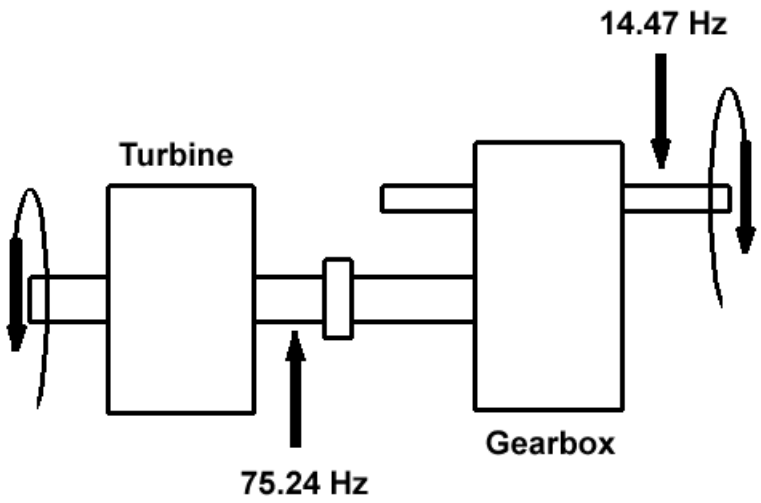


Figure 10

- The turbine illustrated in Figure 10 drives an entire paper machine including the peripheral equipment.
- Note the gearmesh frequency and sidebands at input and output shaft turning speeds in the data. These frequencies make diagnosis slightly more difficult. Although the input shaft sidebands are higher in amplitude, additional spectral data should be collected with improved resolution to separate closely spaced peaks. 5x output shaft sidebands are adding to the input shaft sidebands, making diagnosis more difficult.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The gear mesh frequency in Figure 11 appears at 14xTS of the pinion. Note that this peak is apparent in all three spectra.

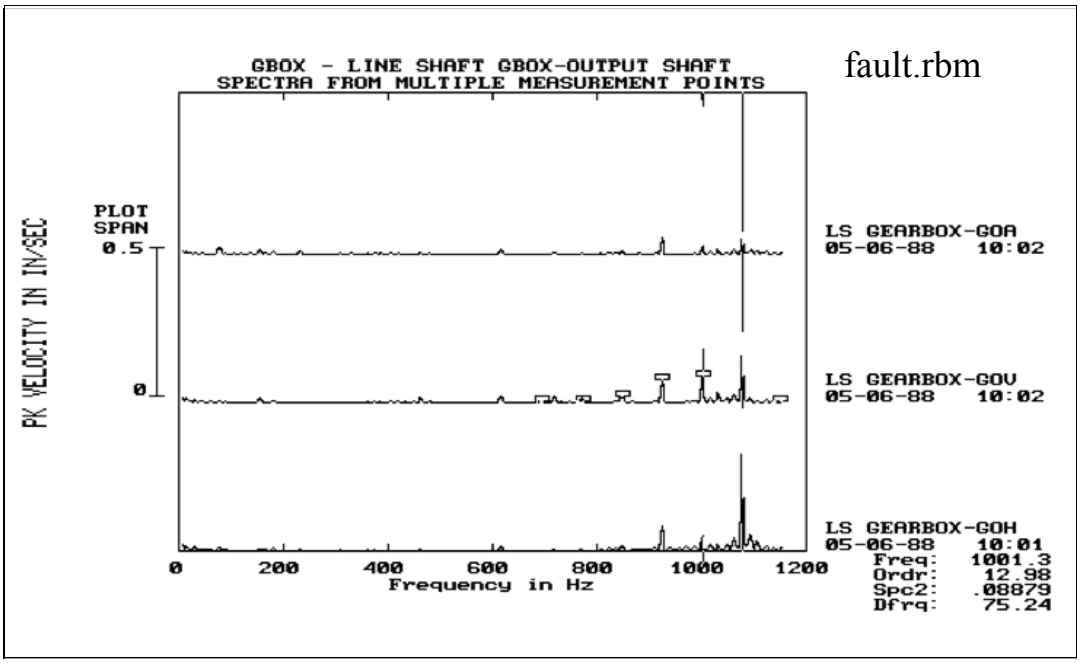


Figure 11

Sidebands provide the key to gearbox analysis. Figure 12 shows sidebands of the gearmesh frequency modulated by 1xTS of the output shaft.

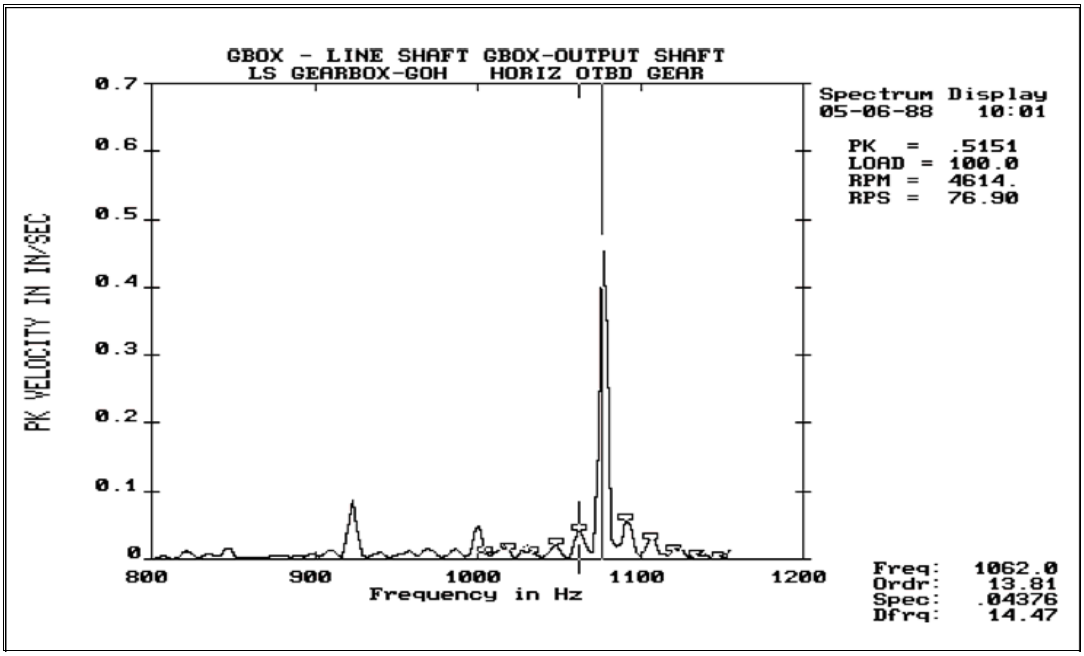


Figure 12

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Sidebands modulated by the speed of the input shaft appear on the spectrum in Figure 13. Because sidebands of the output shaft are also present, it might prove more difficult to determine which shaft has the defect. The input shaft is more suspect because of the higher amplitude of its sidebands.

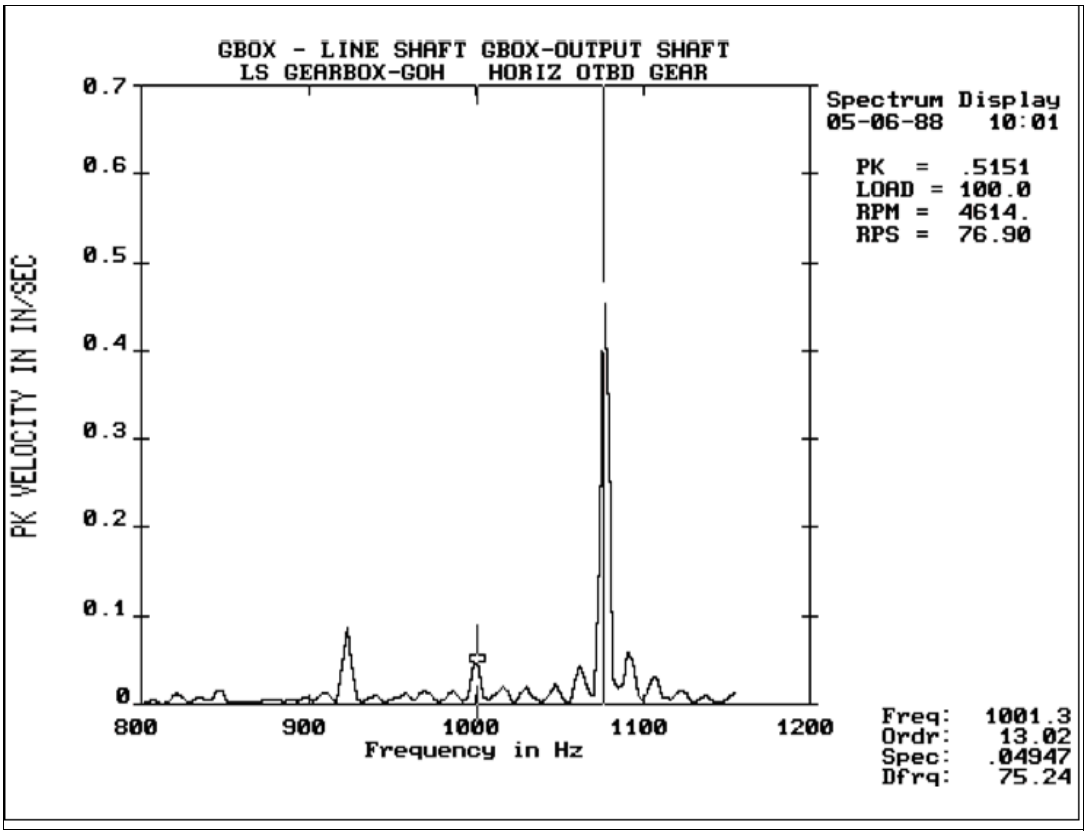


Figure 13

With sidebands of both the input and output shaft speeds, it is very likely that there are defects on both gears. It is common for a defective gear to "share" its defect with the mating gear. Eventually both will need to be replaced or repaired. To avoid introducing more problems, the best solution would be to replace both gears.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #2

Figure 14 is a multi-point plot of the gearbox points. Proper diagnosis requires closer scrutiny

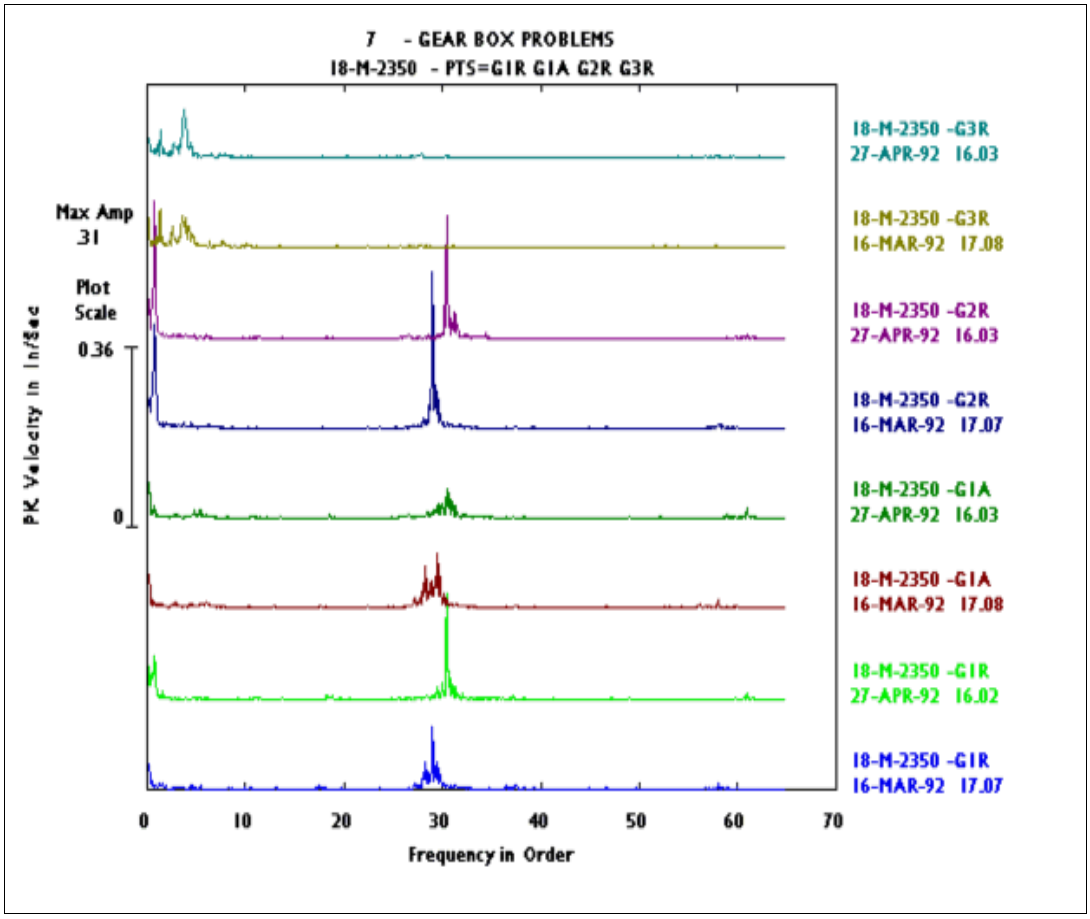


Figure 14

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Examining the single point G2R in Figure 15 helps indicate the presence of a significant amount of high frequency energy. The turning speed of this shaft is 879 rpm. The turning speed of the output shaft at point G3R is 293 rpm. Sidebands are appearing around gear mesh frequency spaced at 0.333 orders of 879, which is 293 rpm. Therefore the shaft rotating at this speed is carrying the defective gear(s).

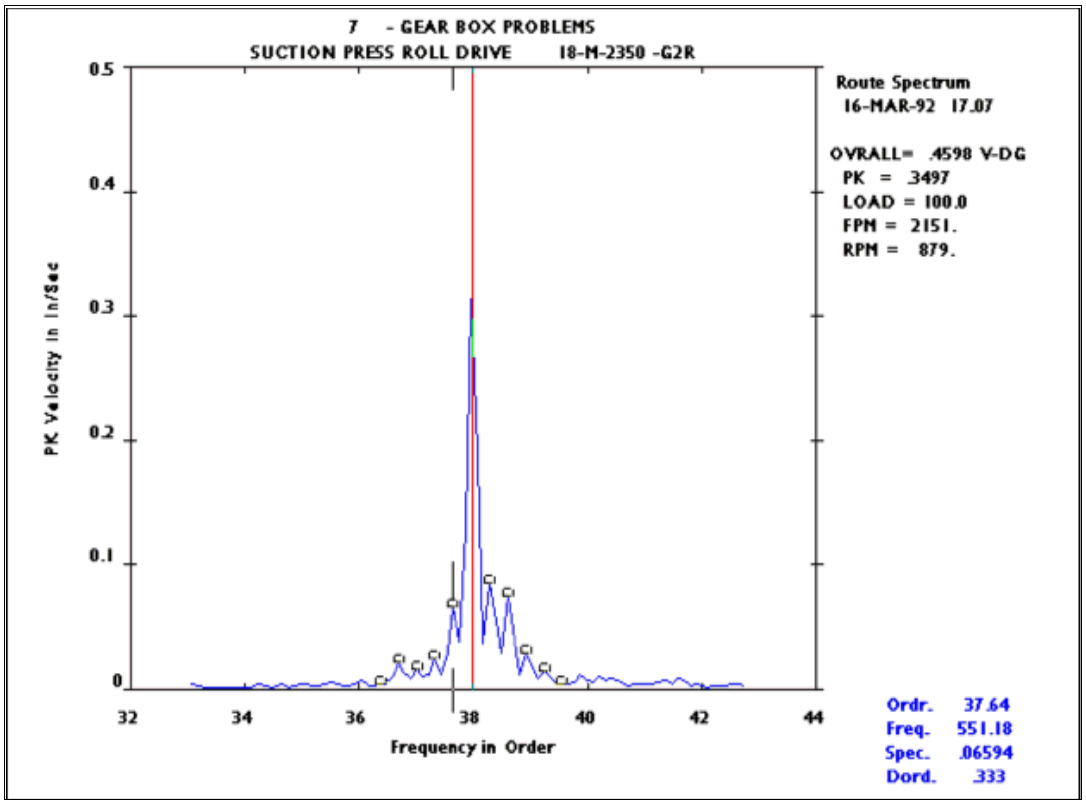


Figure 15

The spacing of the sidebands as indicated by the Dord is 0.333. The turning speed of the shaft from which this plot is taken is 879 rpm, 1/3 of 879 is 293 rpm. In this case the bad gear was on the G3R point, the output shaft.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

This gear box had additional problems. 3xGMF is visible in the spectrum in Figure 16. This is representative of a more advanced stage of gear misalignment and more severe gear problems.

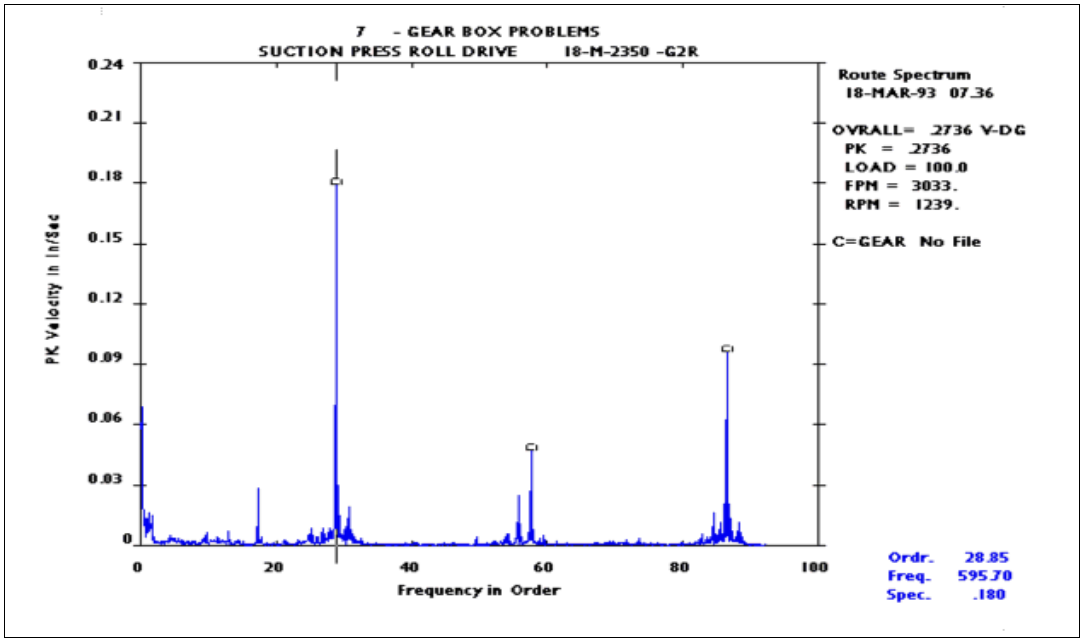


Figure 16

After repairs were made on the gear box the 3xGMF virtually disappears. GMF is still present, but this is normal even on a gear box in good condition.

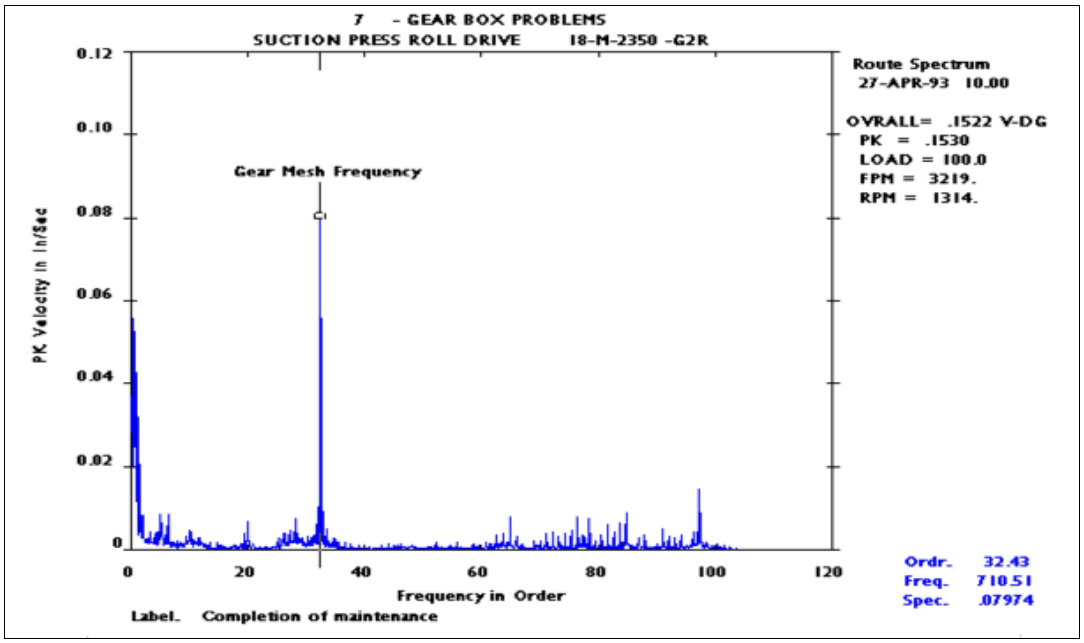


Figure 17

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #3 - F.D. Fan #8

In Figure 18 the cursor is marking the Runspeed of the shaft at 1257 rpm. There are no Fault Frequencies for this example. This makes solving gear box problems more challenging.

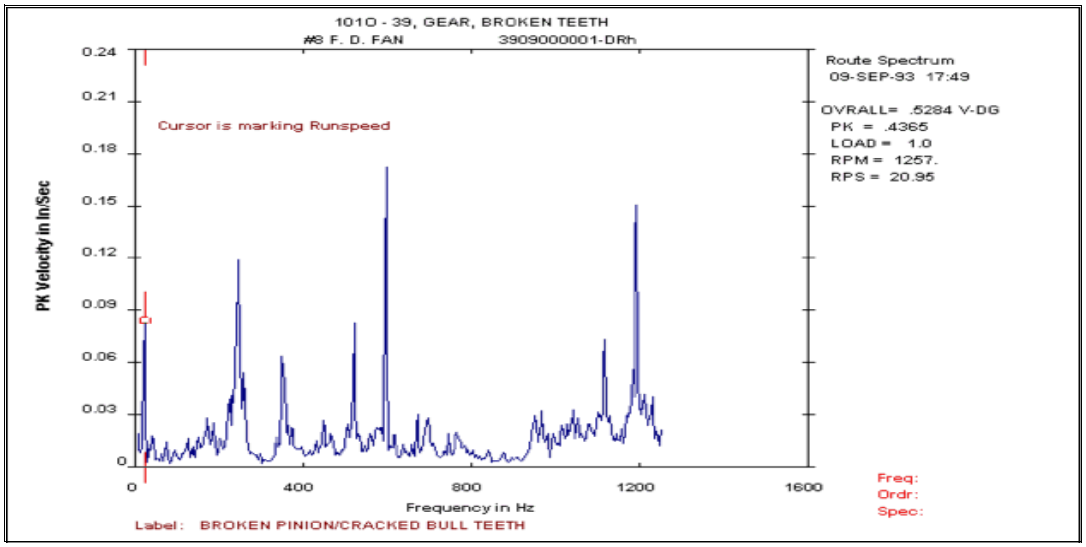


Figure 18

The waveform of this measurement point reveals a very high G swing. This is typical of a gear-box with a gear problem.

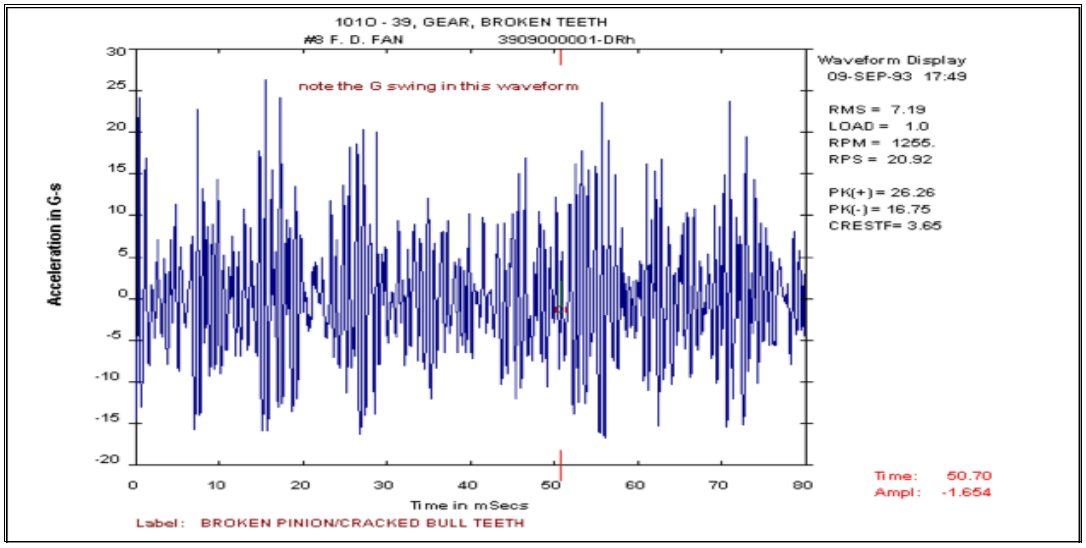


Figure 19

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Figure 20 is a spectrum with gear mesh frequency ***marked*** and sideband cursors marking sidebands evenly spaced at the runspeed of the shaft with the bad gear on it. This spacing is at 74.42 Hertz, which is 4465 rpm. Determine what operates at this frequency. The shaft running 4465 rpm is the shaft with the bad gear on it.

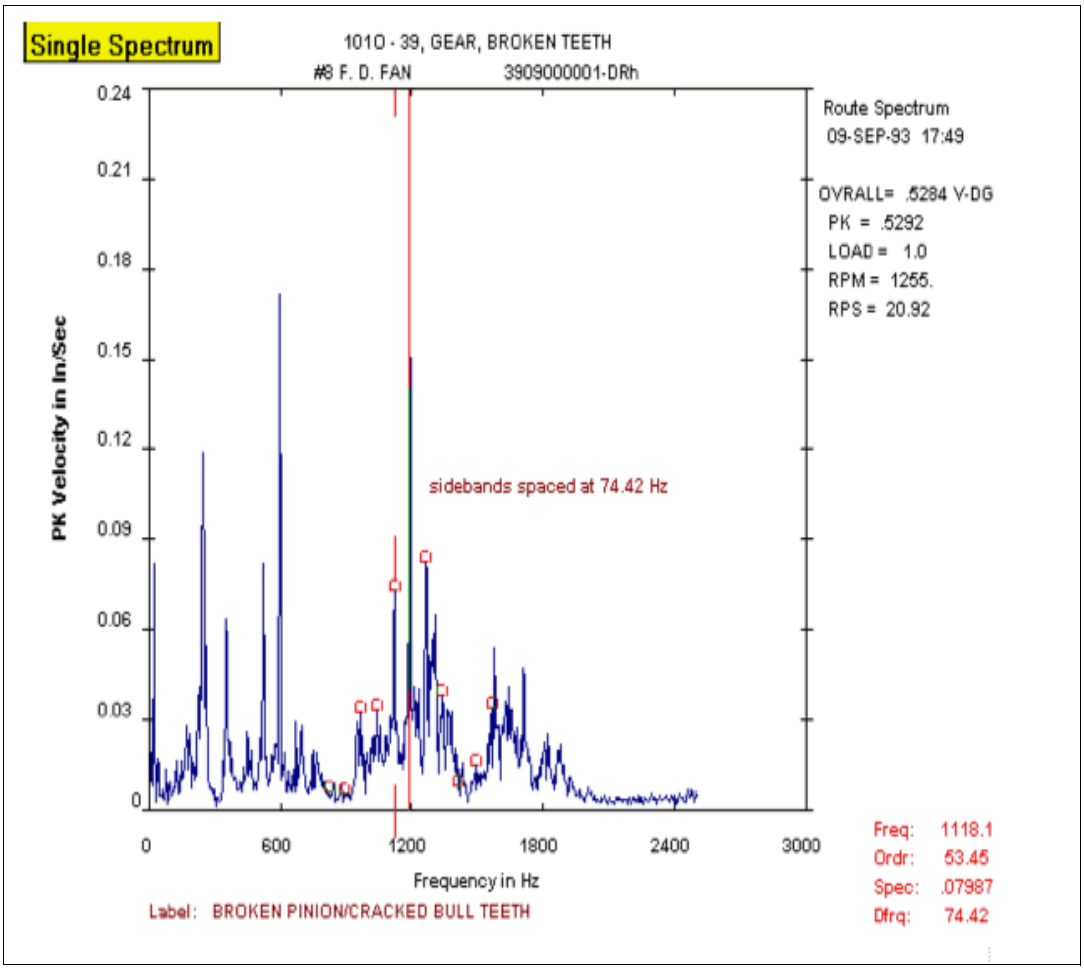


Figure 20

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #4 - Vacuum Pump Gear-Box

Knowing specific gearbox data and operating frequencies makes analyzing data much easier. Fault frequencies aid analysts in determining the fault. 3xGMF is present in the data in Figure 21.

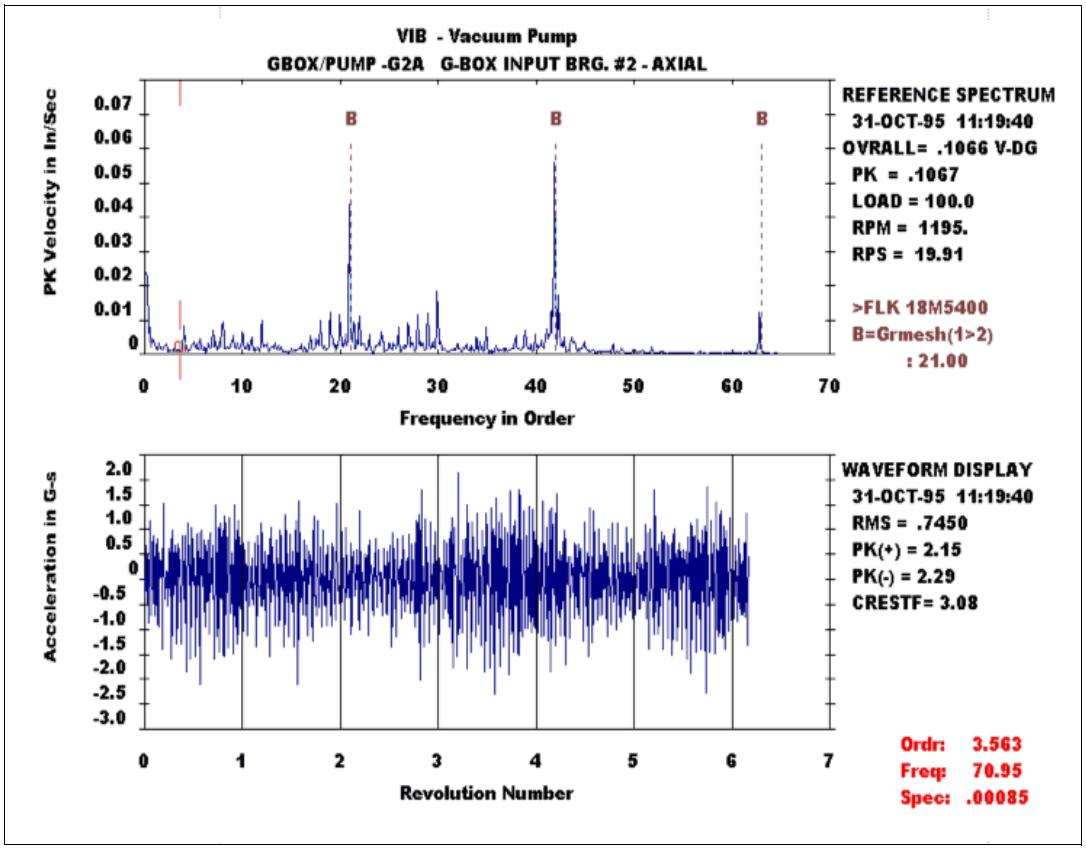


Figure 21

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The single spectrum in Figure 22 provides a good view of gearmesh frequencies and related sidebands.A full scale plot helps to examine the equally spaced peaks at GMF.

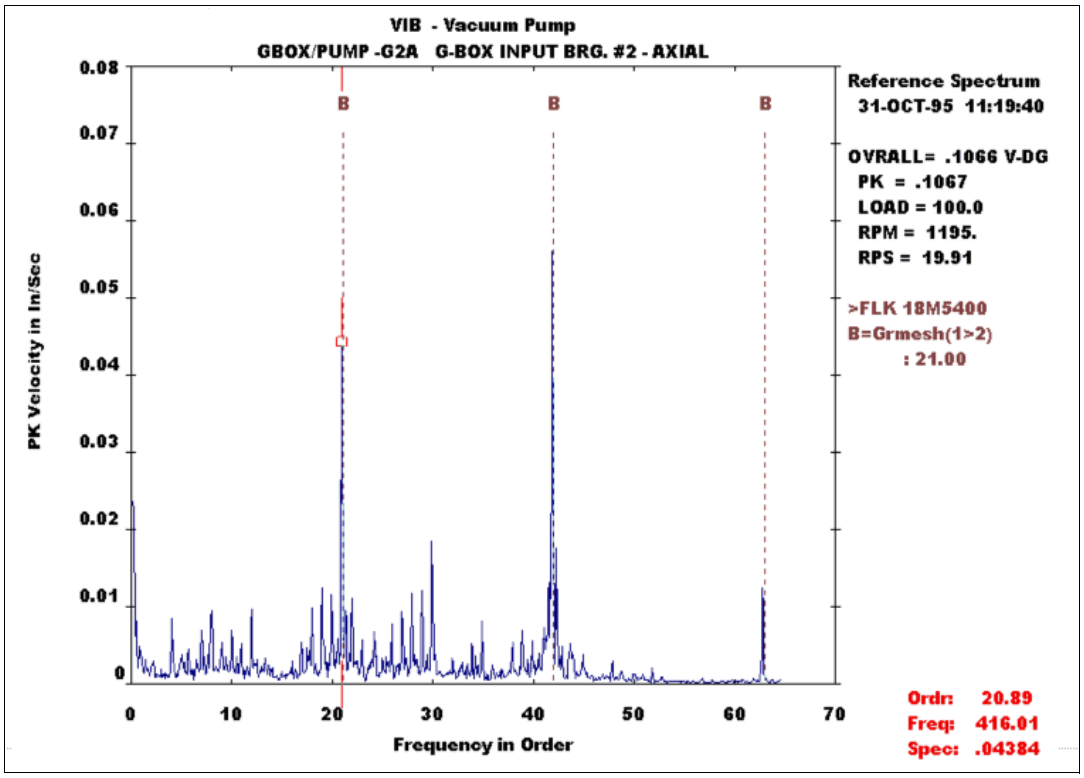


Figure 22

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Gearmesh Frequency is marked. The cursor is moved to the lower sideband. The difference between the two frequencies is 1195 cpm. A family of sidebands are marked with boxes (See Figure 23).

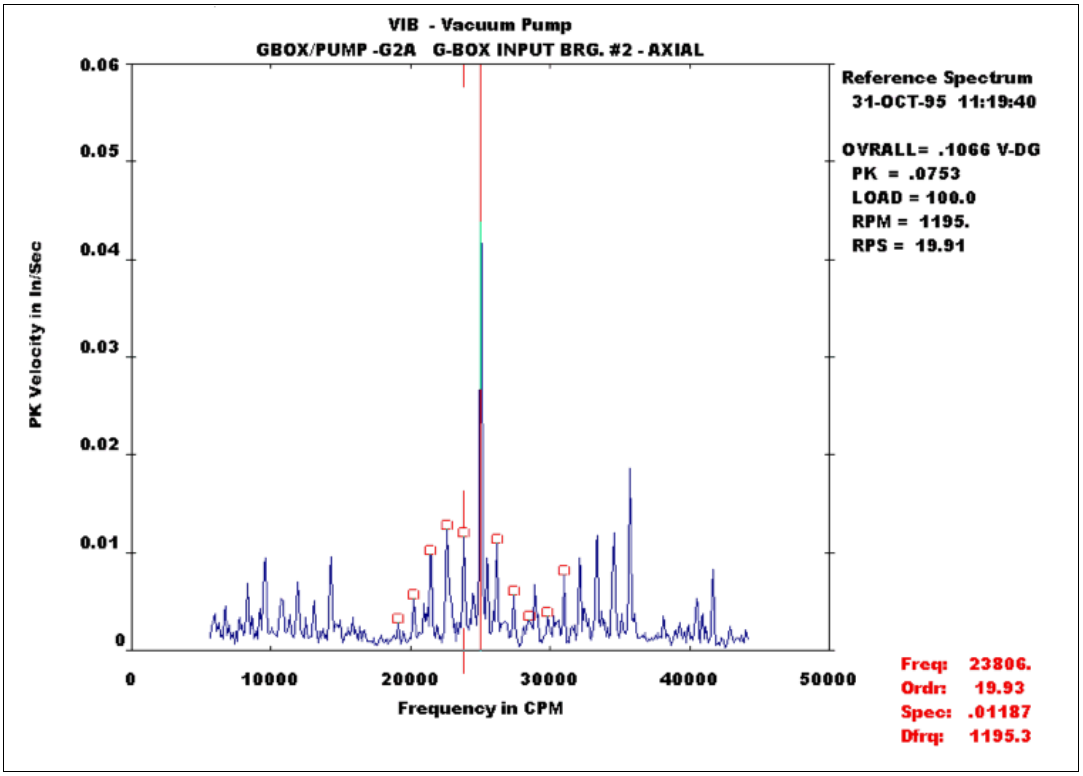


Figure 23

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Helpful information for successful gear box analysis:

1. Simple diagram or sketch of drive-train is essential
2. Input and output shaft speeds
3. Intermediate shaft speeds if any
4. The number of teeth on each gear
5. The type of gears in the gear-box
6. An F_{max} to include 3.5 x GMF
7. Adequate Resolution to separate sidebands
8. The mounting method of the accelerometer must be considered. A stud mounted accelerometer may need to be installed at each shaft to obtain good GMF data. Analysts may need to experiment a little collecting data to get the best results

Failing to follow these guidelines can result in gear-box failures that will add to operating and maintenance costs.

Gear Box manufacturers can be a source of information that you need. Make a list of units for which you need information before you call them.

Opening spare units in storage is another way to get the information.

Take advantage of down time and disassembled units to collect pertinent data.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Belt Defects

Section 7

Objectives

- Calculate frequencies for belt defects.
- Recognize belt defect patterns in spectral and time waveform data.

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

Belt Defects

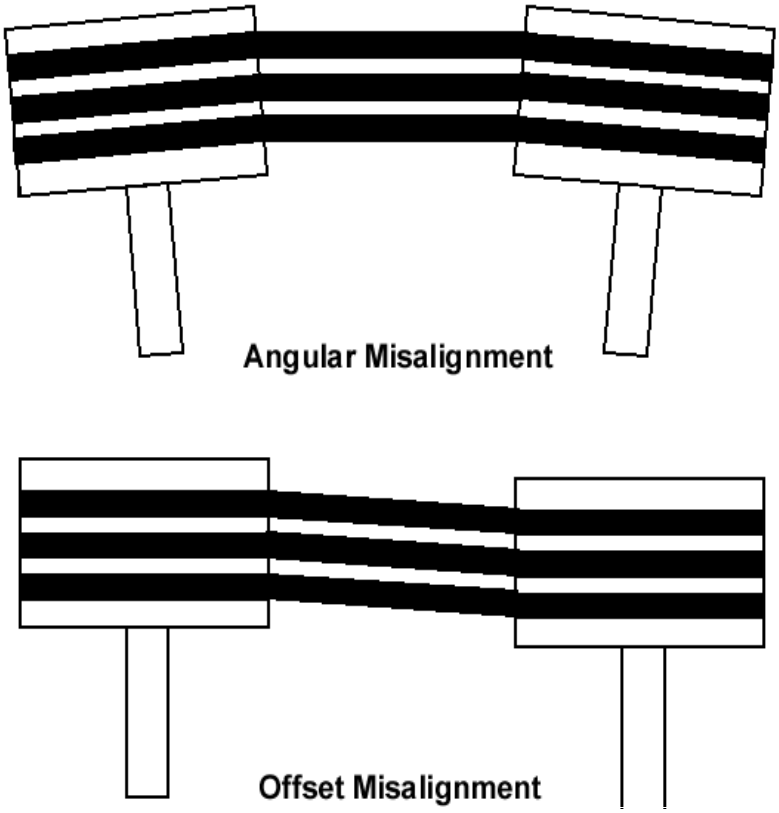


Figure 1

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

Case History #1 - Belt Driven Vacuum Fan

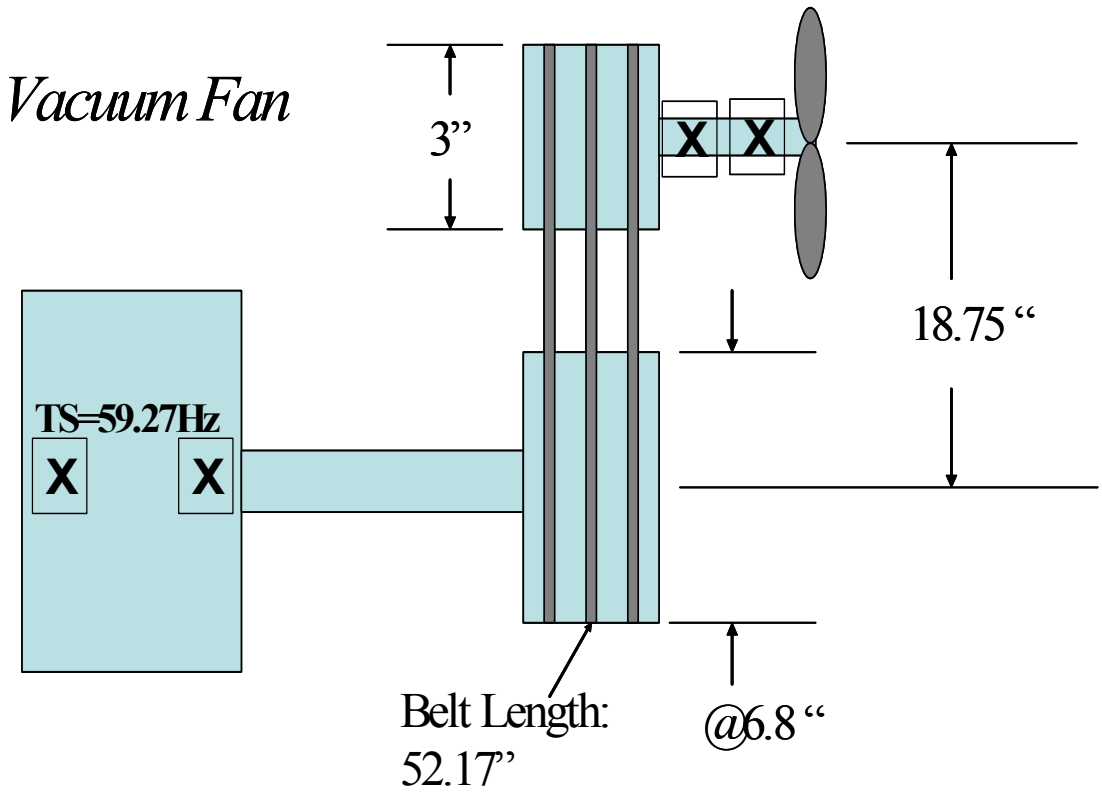


Figure 2

Note
Many software packages can calculate these frequencies. For more accurate frequency calculations, measure diameters and distances within 0.25”.

$$Belt\ Frequency = \frac{\pi \times Sheave\ Speed \times Sheave\ Diameter}{Belt\ Length}$$

Where π = 3.1416

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

Sheave Diameter = Diameter in inches of dirver OR driven sheave

Sheave Speed = Turning speed of driver OR driven sheave

Belt Length = Length of belt in inches found on the belt nameplate or measured

Note

Sheave diameter and sheave speed MUST come from the same source. For example: Diameter of driving sheave and speed of driving sheave.

The fundamental belt frequency for the system in Figure 2 would be:

$$Belt\ Frequency = \frac{3.1416 \times 6.8in \times 59.27Hz}{52.17in}$$

Belt Frequency = 24.27 Hz

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

Case History #1 - Belt Driven Vacuum Fan

The spectrum in Figure 3 comes from the vertical direction on the drive motor. The turning speed of the belt was calculated at approximately 24.27 Hz. Turning speed is the speed at which the belt actually makes a complete cycle around the two sheaves.

The spectrum may show a peak at the primary belt frequency. In this case, the amplitude of the 2 X belt frequency peak is higher, because the defect on the belt impacts both sheaves during each belt revolution.

Motor shaft turning speed is 59.27 Hz. A peak caused by the turning speed of the fan appears at 113 Hz. The vibration caused by the defective belt adds a lot of energy into the system.

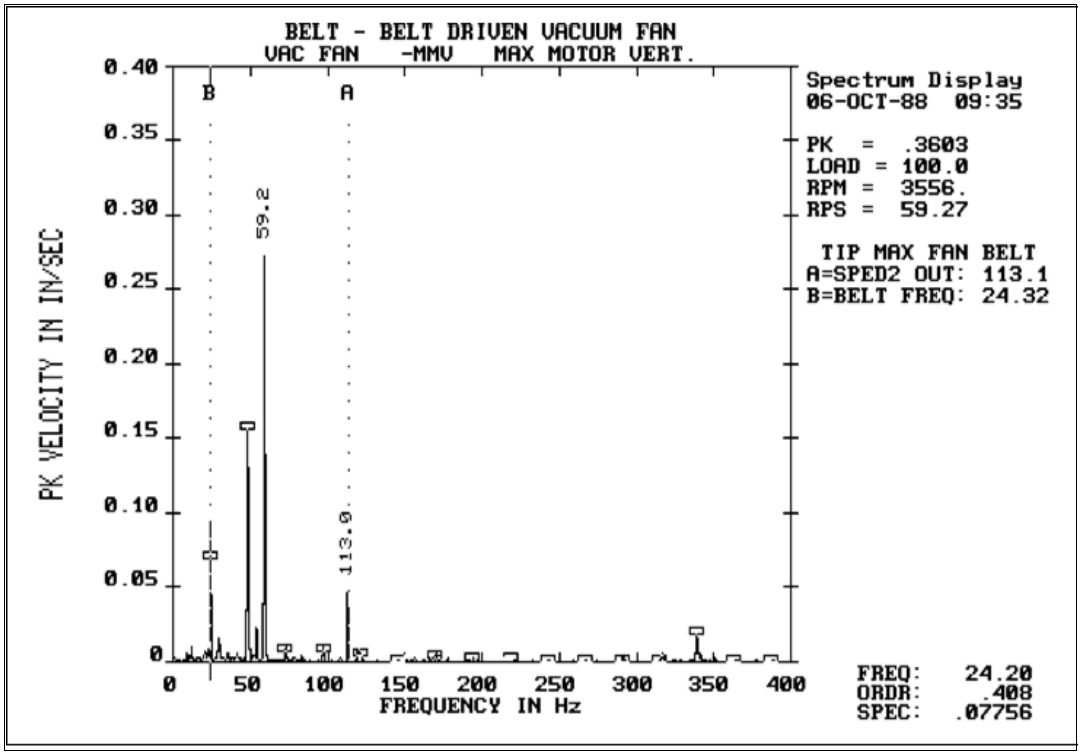


Figure 3

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The spectrum in Figure 4 comes from a permanently mounted transducer located on the **FAN** housing. The plot shows a very high 2 X belt frequency. The 2 X belt frequency shows a peak amplitude similar to the peak caused by the unbalance of the fan.

Resolving the belt problem would eliminate almost half of the energy in the system.

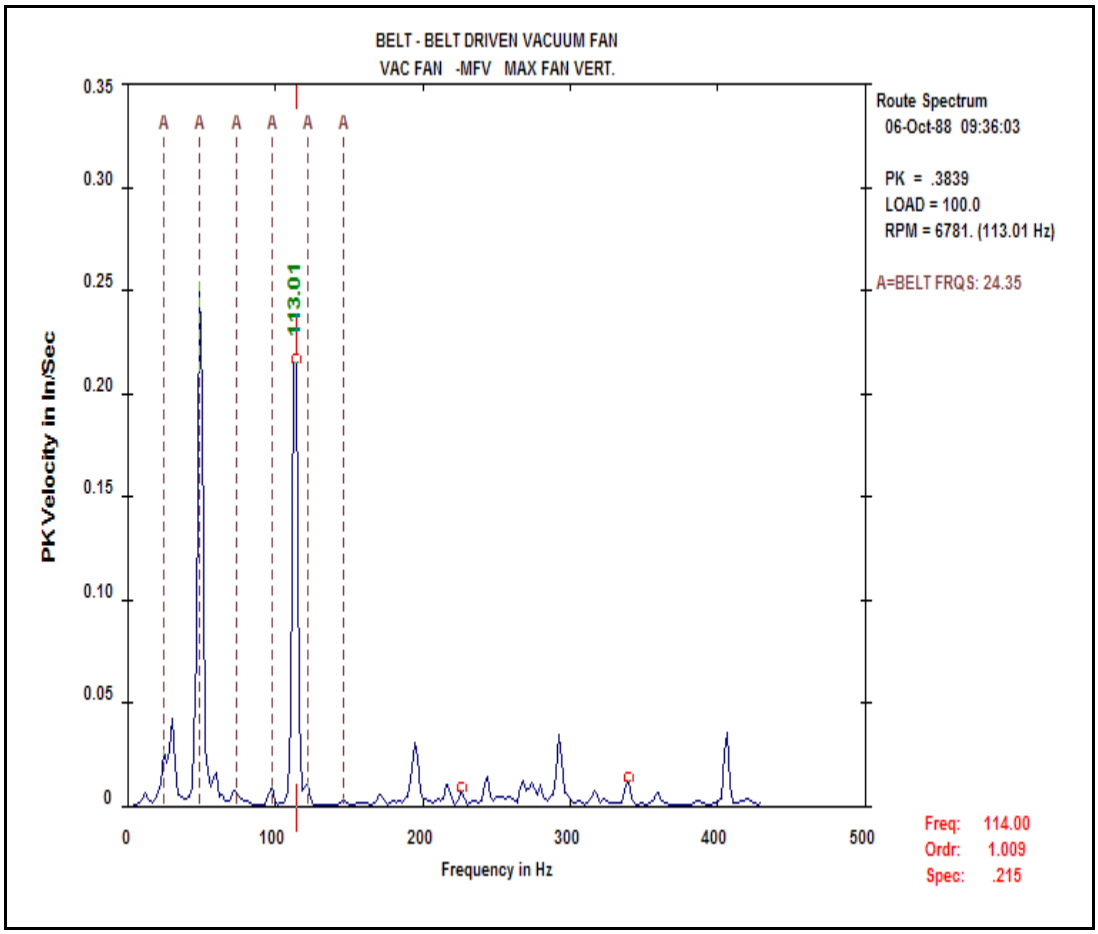


Figure 4

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The time waveform from the motor position shows a modulated pattern of vibration. This type of pattern commonly occurs on belt driven equipment. The large cycle represents the difference in frequency between the turning speed peak on the motor and the 2 X belt frequency peak. (See Figure 5)

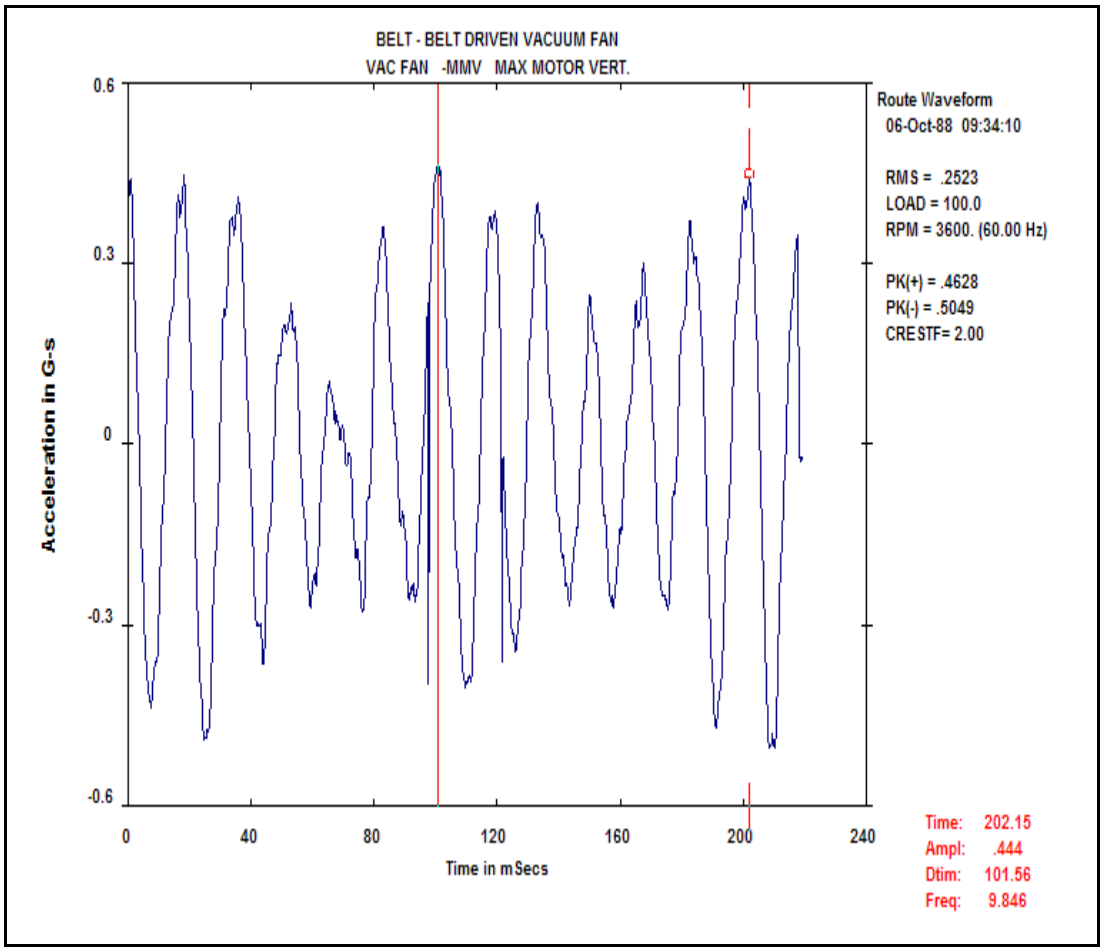


Figure 5

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

Case History #2 - Forced Draft Fans

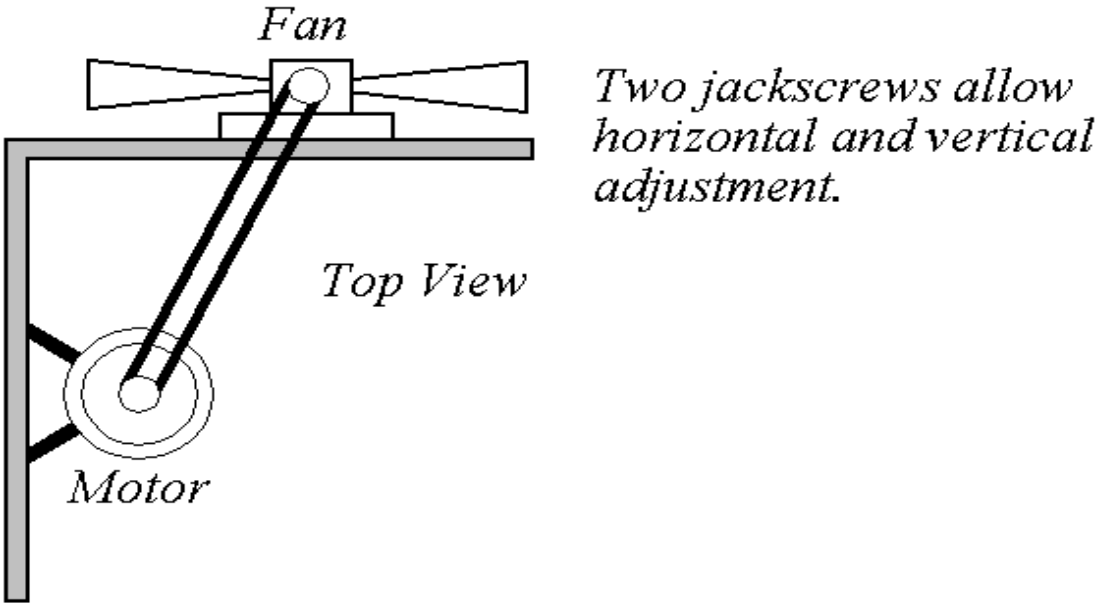


Figure 6

Equipment Information

◆ Fourteen of these fans dried coffee beans treated with a decaffeinating agent.
◆ Note the absence of gusseting support in the diagram.
◆ The spectral data collected on the motor showed a higher amplitude at the fan turning speed frequency than at the motor turning speed frequency.
◆ The spectral data collected on the fan showed a higher amplitude at the motor turning speed frequency than at the fan turning speed frequency.
◆ Impact testing revealed that motor, fan, and 1 X belt frequencies could all excite resonance.

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

A multiple-point plot of all motor points is shown in Figure 7. Very little significant high-frequency energy appears. Almost all the energy exists in humps at low frequencies.

These spectra come from a newly installed machine. Therefore, the low-frequency peaks show cause for concern even though their amplitudes remain low. The dominance of the axial reading is also unusual.

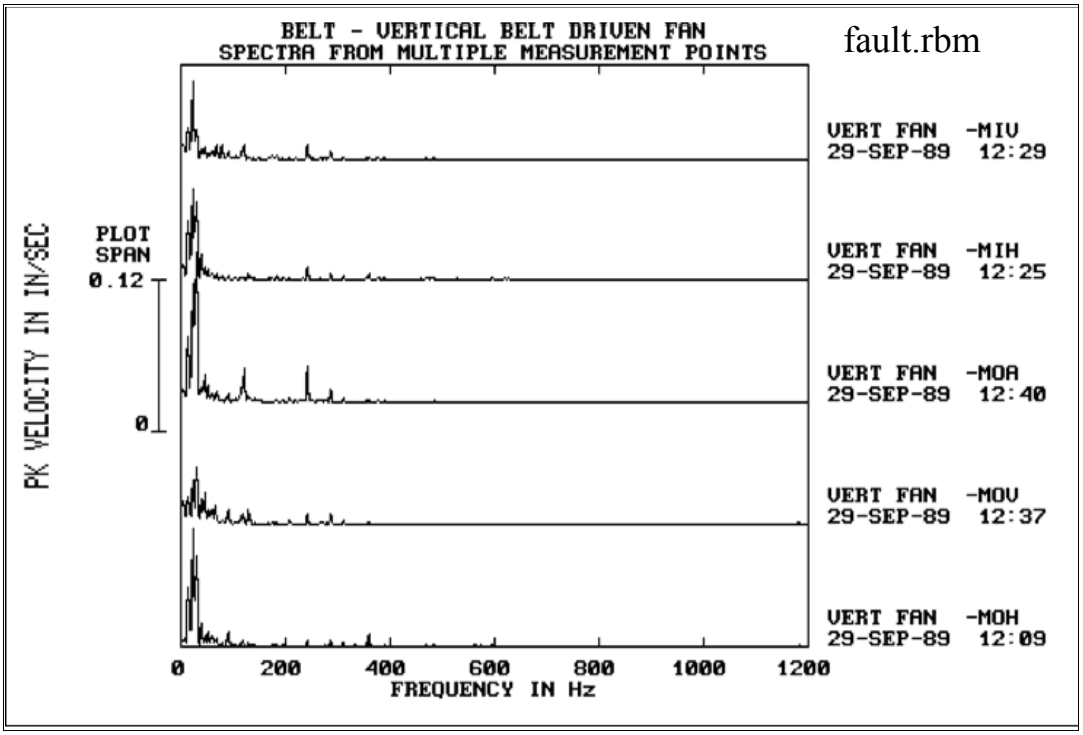


Figure 7

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

In Figure 8, the motor points spectra are expanded to help analyze low frequencies. Each individual, low-frequency peak is evident. The turning speed of the belt causes the 13.2 Hz (792 RPM) peak. The 22.2 Hz (1332 RPM) peak results from the turning speed of the fan. The 29.5 Hz (1770 RPM) peak comes from the turning speed of the motor.

Note that the amplitude of the turning speed peak of the fan exceeds that for the motor in spite of the fact that the measurement was made on the motor. Note also the level of the axial vibration. Axial vibration can indicate misalignment between the motor and fan.

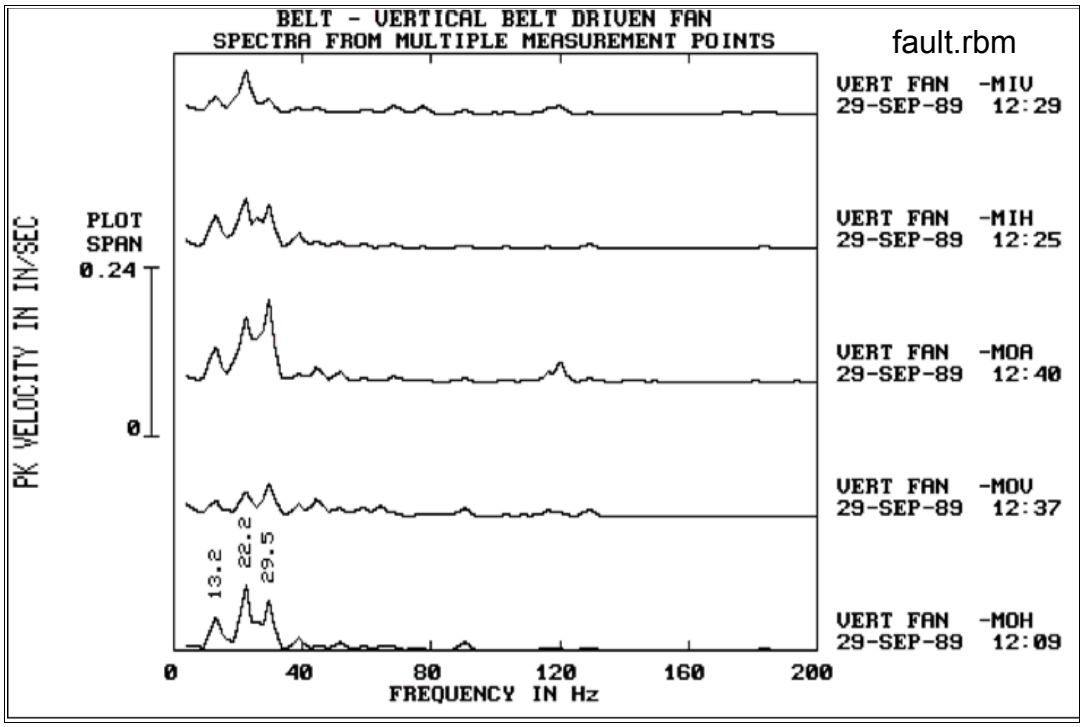


Figure 8

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

A multiple-spectra plot for all the fan points is shown in Figure 9. Significant peaks show up around 200 Hz, particularly on the fan outboard axial point, FOA. Most of the remaining energy occurs in the low-frequency humps. The amplitudes remain low for all points, but the pattern of the peaks indicates that an installation or design problem might exist.

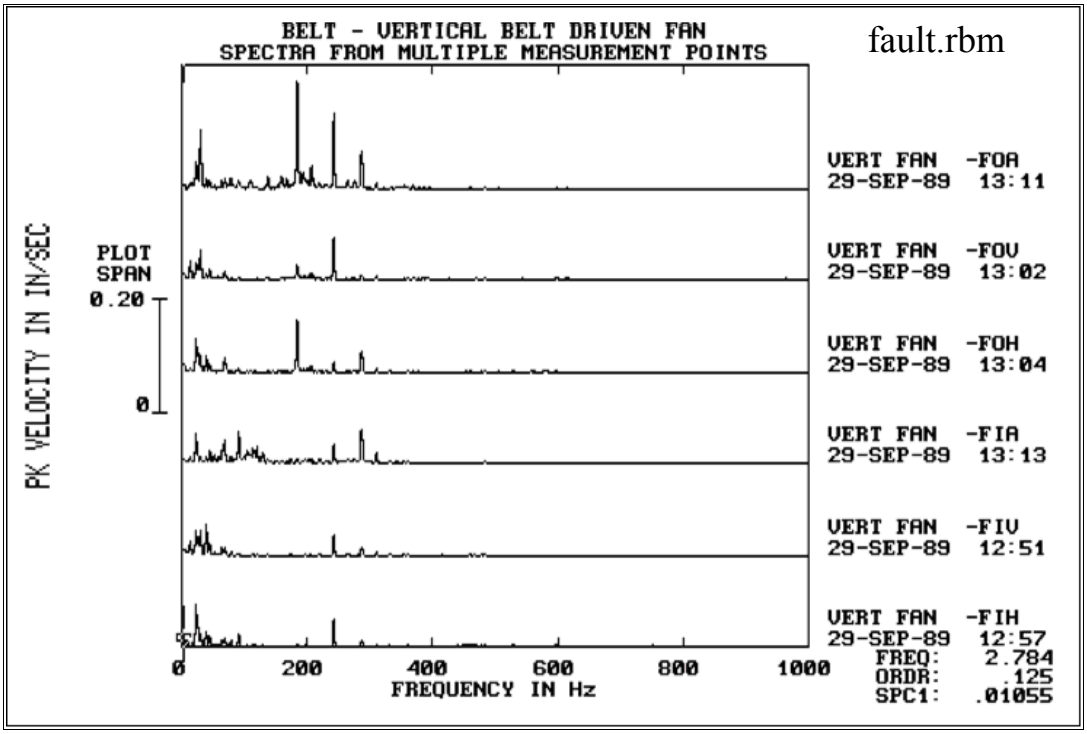


Figure 9

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The full scale plot in Figure 10 is from fan inboard vertical point, FIV. A vertical mark appears on the blade pass frequency peak at 11xTS. The sideband cursor appears at 12xTS. This plot shows sidebands of fan turning speed around the blade pass frequency.

The sidebands may result from fluctuations in the turning speed of the fan. These fluctuations may result from improper tensioning of the belts during installation. A lot of energy still shows up at the turning speeds of the motor, fan, and belts, but most remains below 50 Hz.

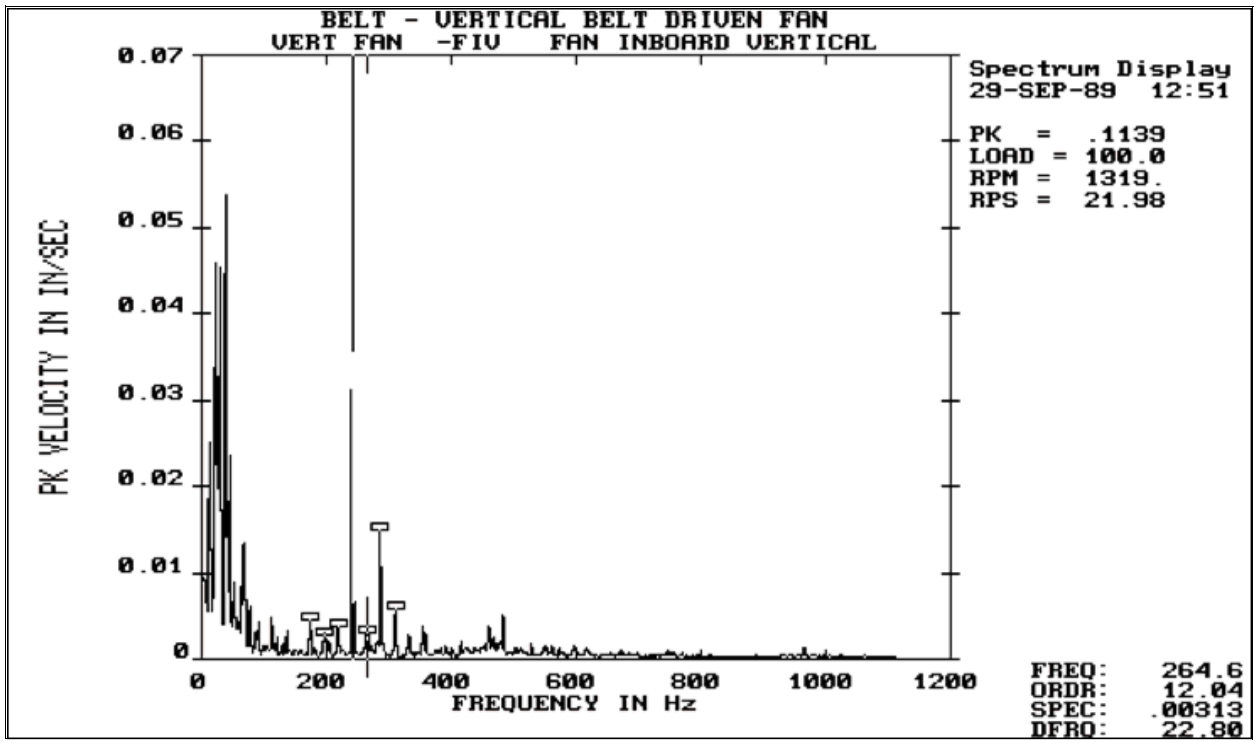


Figure 10

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The plot in Figure 11expands the lower 200 Hz of the FIV data for further analysis. The belt turning speed is approximately 13.1 Hz (786 RPM). Fault frequency lines and cursor markers denote harmonics of the belt speed. The turning speed peak of the motor is 29.5 Hz (1770 RPM). The fan turning speed peak is 22.3 Hz (1338 RPM). The 3 X belt frequency peak at 38.7 Hz is actually higher in amplitude than either the motor or fan turning speed peaks.

Balancing the fan would do little to lower the overall vibration of this unit. The belts probably were not tensioned correctly when installed, so they may be slipping. The high axial measurements either indicate motor and fan misalignment or the presence of excessive run out on at least one of the sheaves.

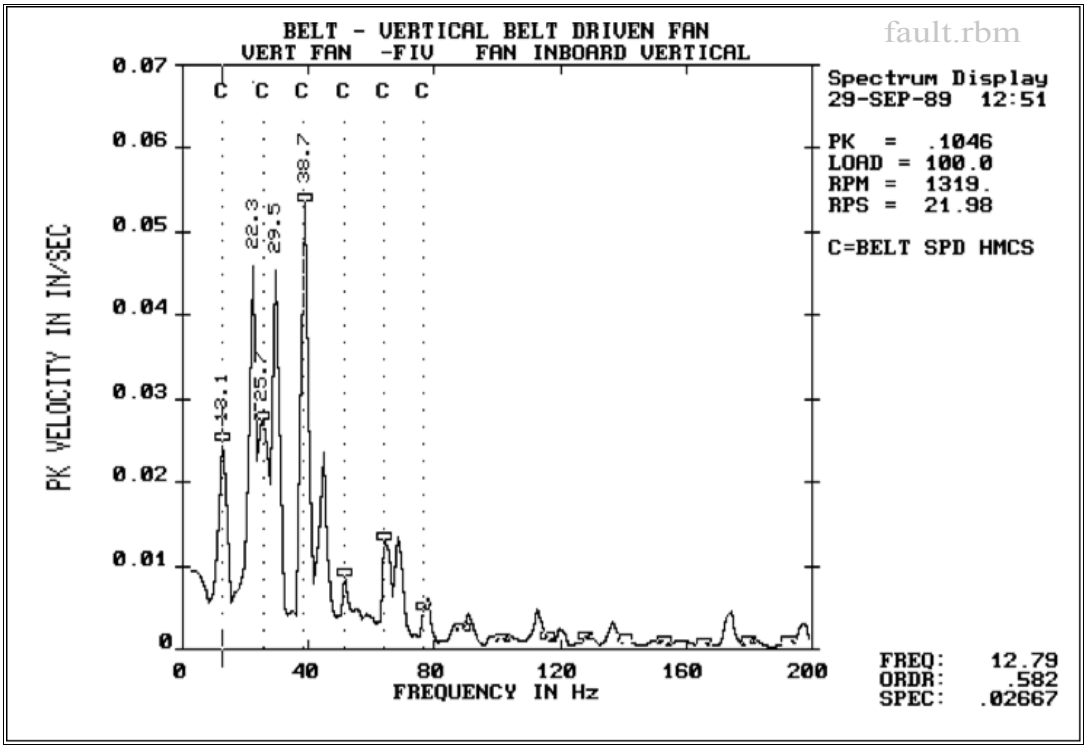


Figure 11

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The time waveform associated with the fan inboard vertical position appears erratic and nonperiodic. Some modulation appears in the signal, but it does not repeat. Evidence points toward a looseness problem. (See Figure 12)

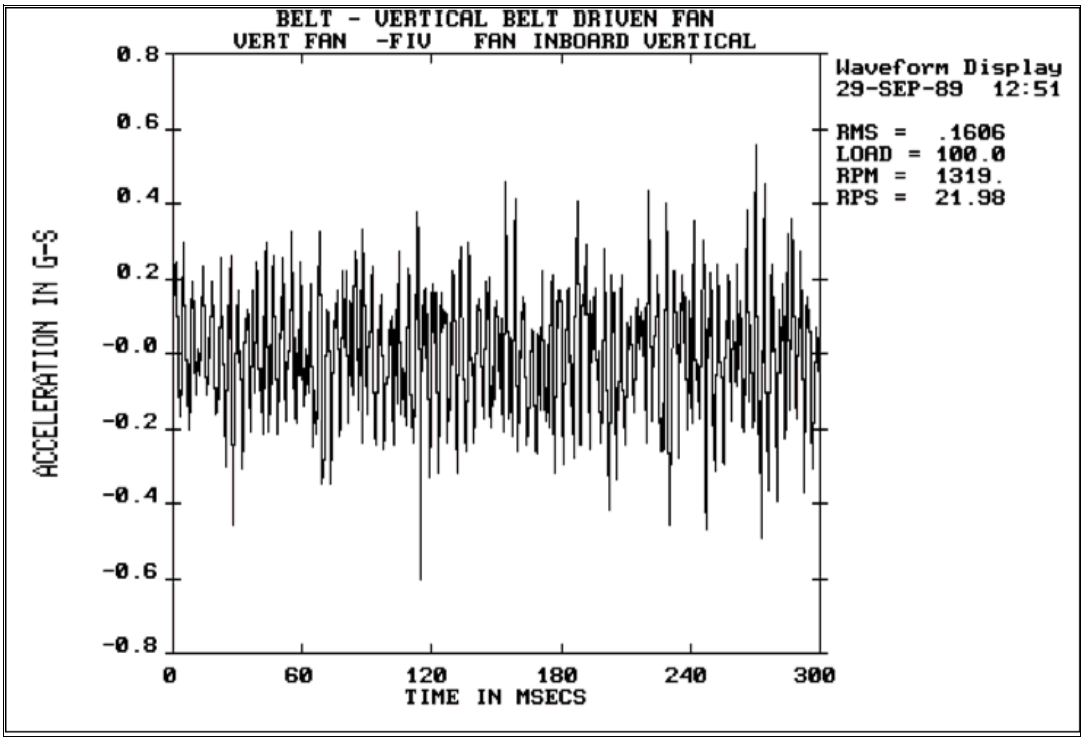


Figure 12

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

Case History #3 - Belt driven over-hung fan

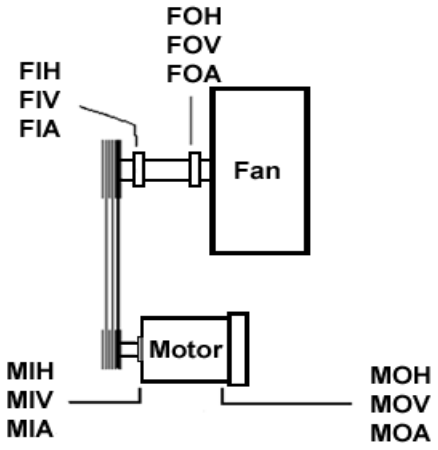


Figure 13

This is a good example of data being collected with inadequate resolution. This data was collected using 800 LOR. Better resolution would separate the peaks in the spectrum. Note the broad skirt (base) on the peak. This is the MOV measurement point. The cursor is marking the turning speed of the motor.

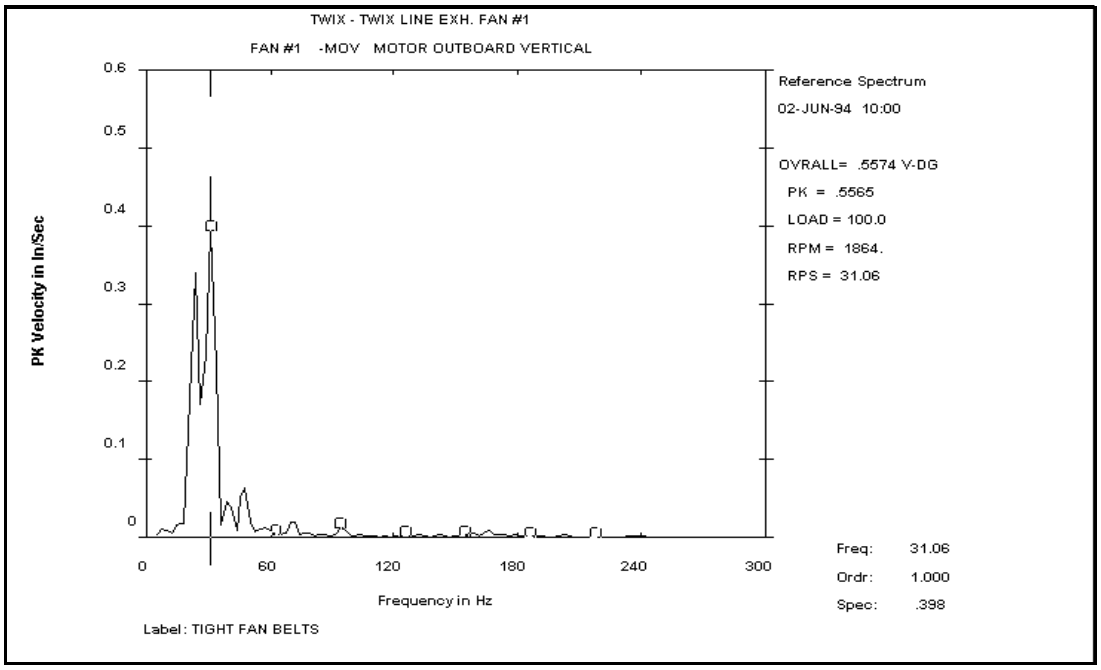


Figure 14

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The spectrum in Figure 15 is from motor outboard vertical, MOV. The cursor is marking the turning speed of the fan. Excessive belt tension is usually indicated by the presence of driver and driven shaft turning speeds appearing in the data.

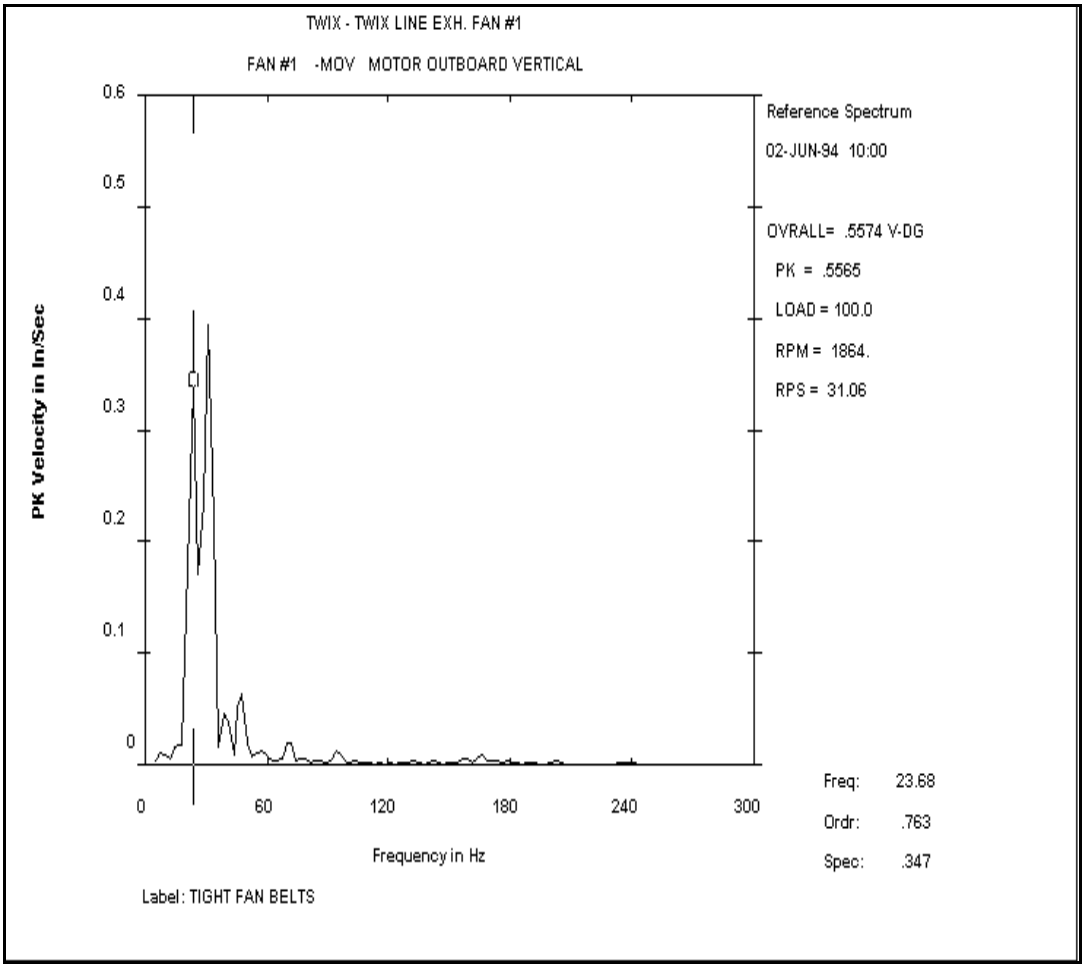


Figure 15

 Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The spectrum in Figure 16 is of the FIV measurement point. The primary cursor is marking the turning speed of the fan at 1423 rpm. The harmonics are of the Fan Turning Speed. This is also an indication of excessive belt tension.

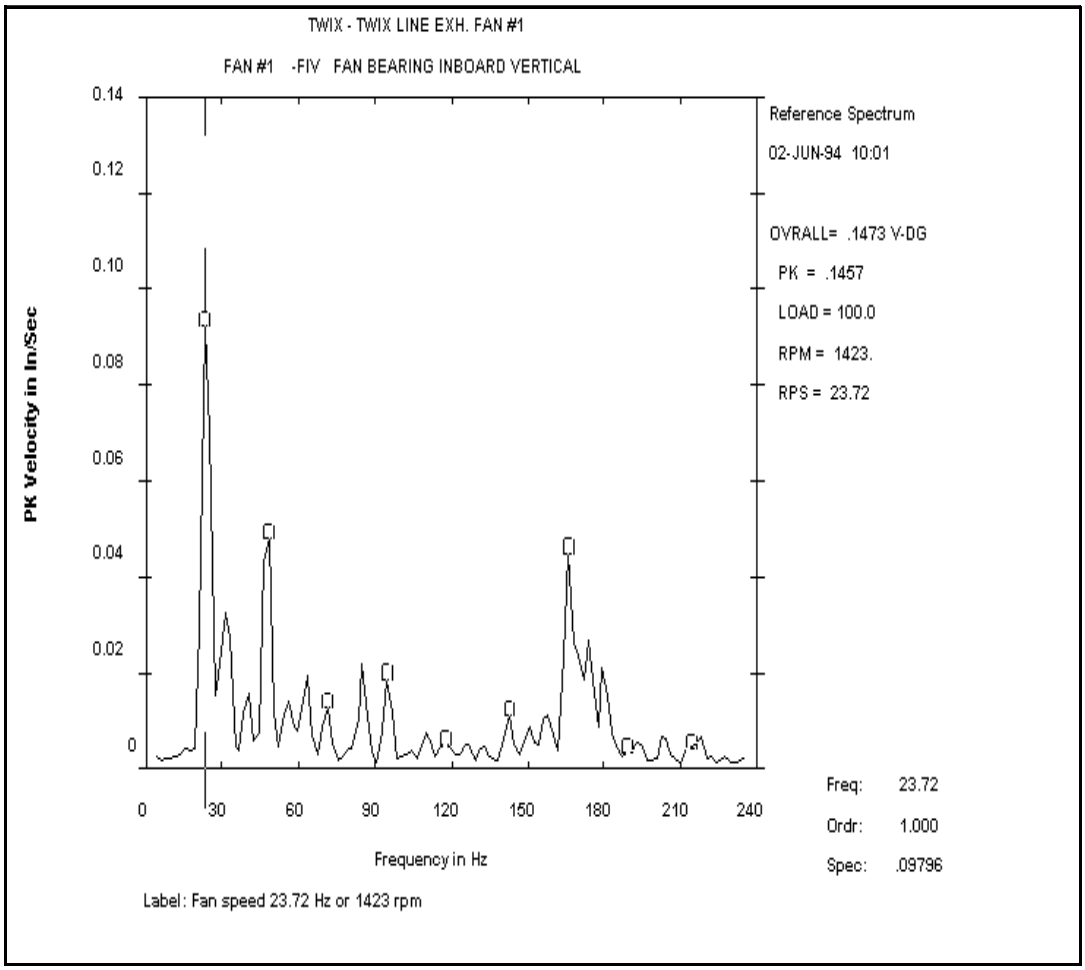


Figure 16

Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

The full scale spectrum in Figure 17 of the FOH measurement point reveals peaks at higher frequencies. The belts being too tight could have loaded the bearings and small faults are starting to appear on the races. Correct tensioning of the belts now could extend the life of the bearings.

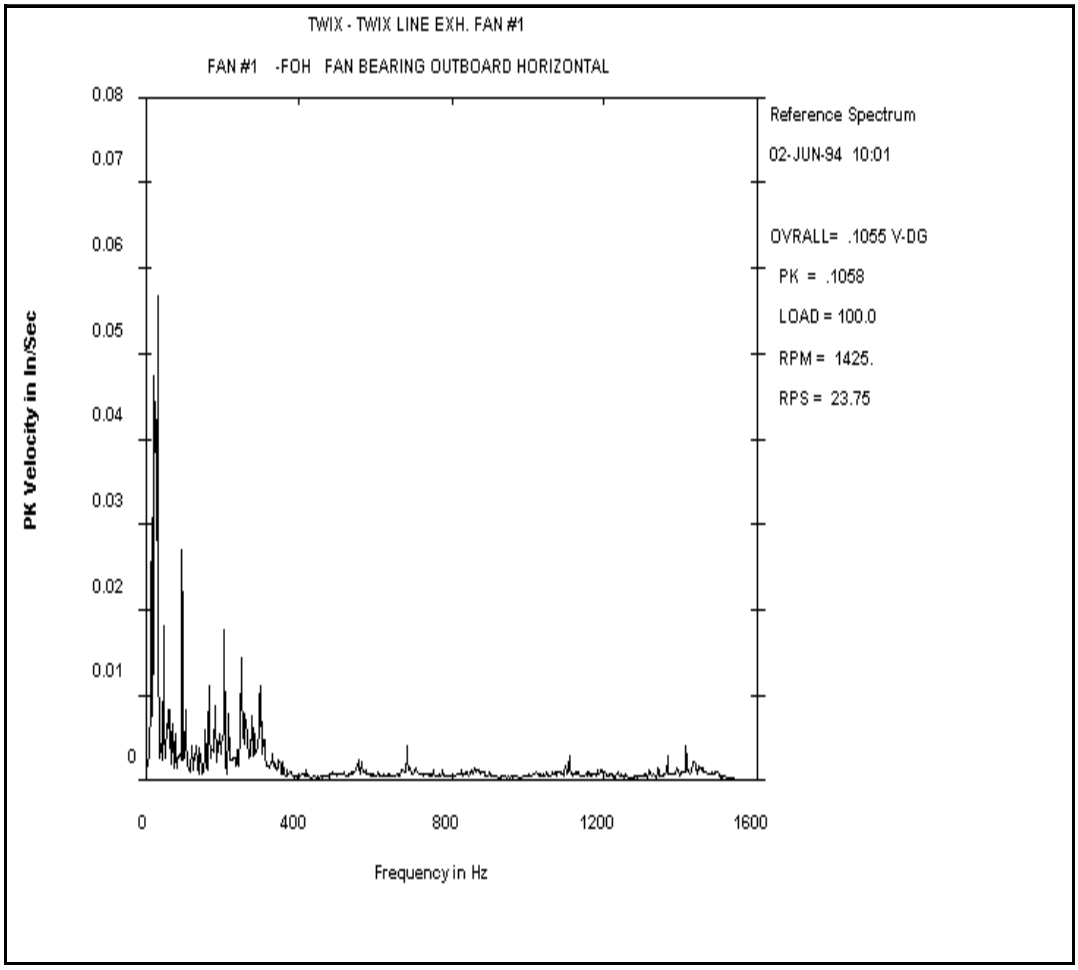


Figure 17

 Copyright 2006, Emerson Process Managment. All rights reserved. Rev 04/06

Electrical Faults

Section 8

Objective

- Review the basic construction of 3-phase induction motors.
- Define electrical motor problems.
- Determine some causes of problems in motors.
- Identify the characteristics of electrical problems in vibration data.
- Understand the association between mechanical and electrical anomalies.
- Determine some corrective measures.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Basic Electric Motor Contruction

There are four fundamental components in an electric motor. A detailed photograph illustrates other parts that make up a typical motor in Figure 1.

- The Stator (the stationary component)
- The Rotor (the rotating component)
- Two "End Bells" or "End Shields" or "Brackets"

These devices are the support mechanisms for the rotor and bearings

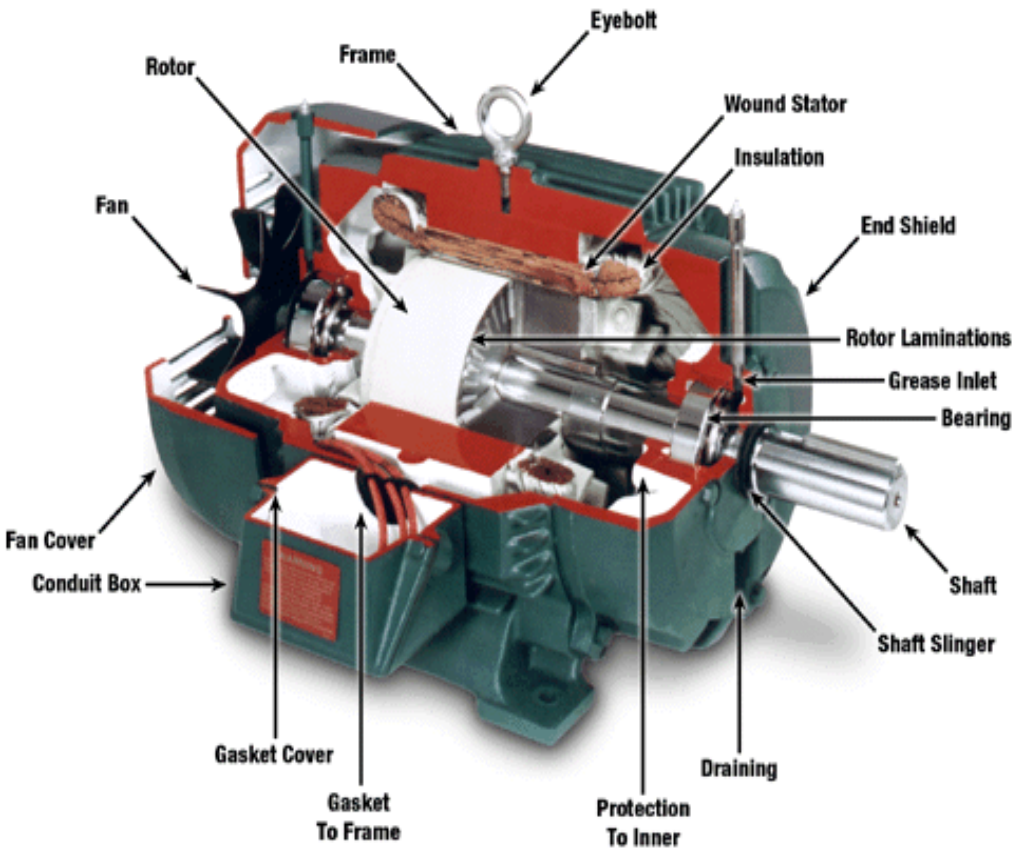


Figure 1

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Figure 2 illustrates the construction of the stator. The rotating electromagnetic field is created in the stator coils. The current field is induced into the rotor, causing the rotor to turn.

Figure 2

A typical squirrel cage rotor construction is illustrated in Figure 3.

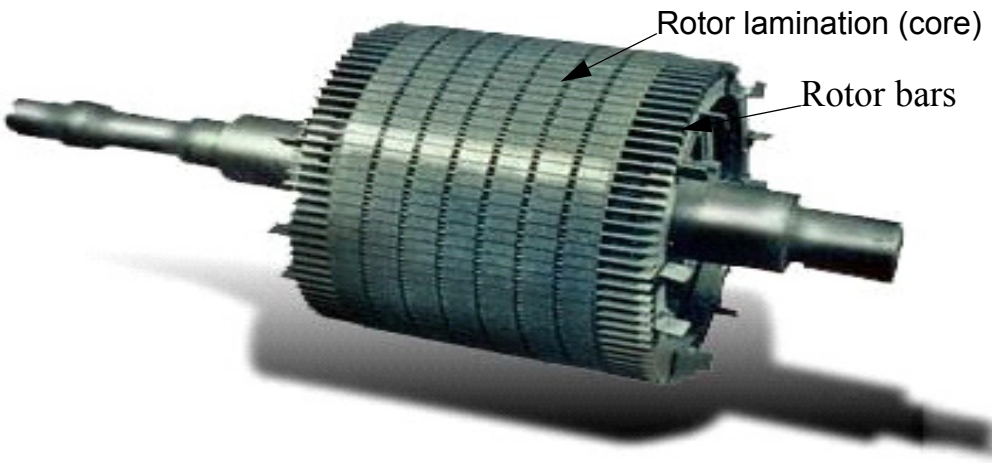


Figure 3

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The configuration and connection(s) in the windings determine the synchronous speed of the motor. An illustration of a simple 3 phase, 2-pole motor is shown in Figure 4.

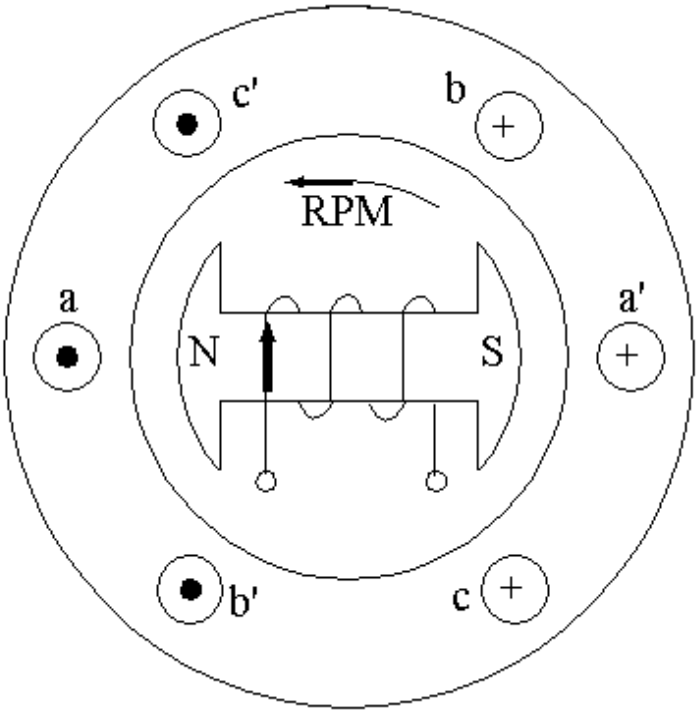


Figure 4

Electric motors can experience many of the mechanical problems discussed earlier such as imbalance, misalignment, looseness, eccentricity, and bearing defects. Each mechanical problem generates a certain vibratory signature. Vibration transducers can detect most of these electro-mechanical defects.

Pure electrical defects are due to disturbances to the electro-magnetic field. These defects also generate distinguishing vibratory characteristics. Uneven or unequal electro-magnetic forces act on the stator or rotor causing vibration. When properly configured, vibration transducers can detect these vibratory signatures.

 Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Other useful tools, parameters and technologies for monitoring the condition of electric motors are:

- Current
- Flux Coil
- Infrared Thermography
- Temperature
- Ultrasonic
- Off line tests

When an electrical problem is suspected, monitor the vibration spectrum in the radial direction and shut off the power supply. If the 2 x Line Frequency ($2xF_L$) signal disappears, an electrical defect is highly probable. In motors that are operating with the rotor being drawn away from magnetic center the vibratory energy will be higher in the axial direction.

Some sources of excessive rotor related vibration are:

- Open bars or cracked end ring
  - Dominant vibration - 1xTS with sidebands spaced at poles X slip frequency
  - Eccentric rotor
  - Dominant vibration - Elevated 1xTS with sidebands spaced at slip frequency and / or $1xF_L$ or $2xF_L$.

Some sources of excessive stator related vibration are:

- Loose stator lamination
  - Dominant vibration - $2xF_L$ with harmonics
- Open or shorted windings
  - Dominant vibration - $2xF_L$ with an increase in amplitude and temperature
- Insulation breakdown
  - Dominant vibration - $2xF_L$
- Phase unbalance
  - Dominant vibration - $2xF_L$

Where F_L = Line Frequency

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Some important frequencies to know when diagnosing a potential electrical problem in induction motors are found in Table 1.

Frequency	Equation
Rotor Bar Pass Frequency	Number of rotor bars x turning speed
Stator Slot Pass Frequency	Number of stator slots x turning speed
Slip Frequency	Synchronous Speed minus Rotor Speed
Number of Poles	$2F_L$ / synchronous speed
Synchronous Speed	$2F_L$ / Number of Poles
Poles x Slip Frequency	Number of Poles x the slip frequency

Table 1

A typical motor nameplate may contain information similar to the one in Figure 5.

Figure 5

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Rotor Defects

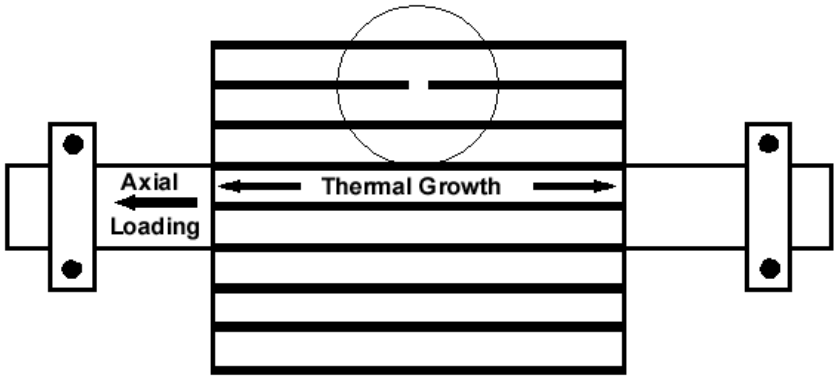


Figure 6

Electrical defects lead to mechanical problems in motors. The rotor bars on both sides of the broken bar must carry more current to maintain motor speed. This condition causes the rotor to have hot spots and heat up unevenly. Besides making the rotor bow, this heat causes the rotor to lengthen. If the bearings do not float properly, this extra length results in excess axial loading on the bearings. (See Figure 6)

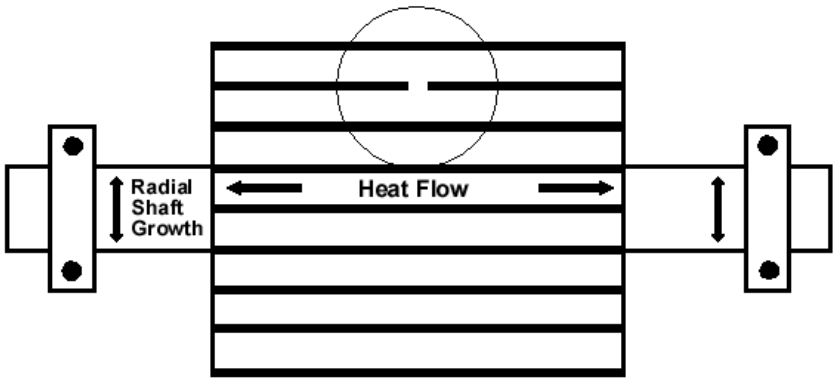


Figure 7

The rotor and shaft heat up excessively over time because of the bad rotor bar. This heat causes axial and radial growth of the shaft. Most of the radial growth of the shaft in the bearings goes toward decreasing the internal clearance of the bearings. If the clearance becomes too small, the bearings overheat and fail. (See Figure 7)

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #1 - Electrical Problem

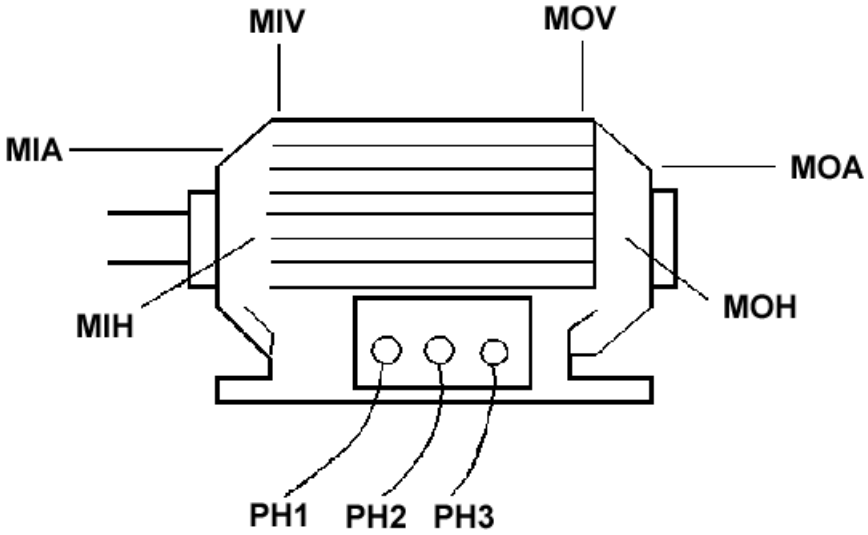


Figure 8

Motor Data:

Horsepower	50
Line Frequency	60 Hz
Turning Speed	59.54 Hz or 3572 rpm

Determine:

Number of Poles = _____

Synchronous Speed = _____

Slip Frequency = _____

Two times Turning Speed = _____

2FL = _____

Poles x Slip Frequency = _____

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Electrical faults can be a challenge to diagnose without adequate resolution. The data in Figure 9 was taken on an electric motor with a potential defect.

Table 1 defines the resolution for each spectrum.

The 400 line spectrum is the regular route data. The other spectra are a product of additional data collection

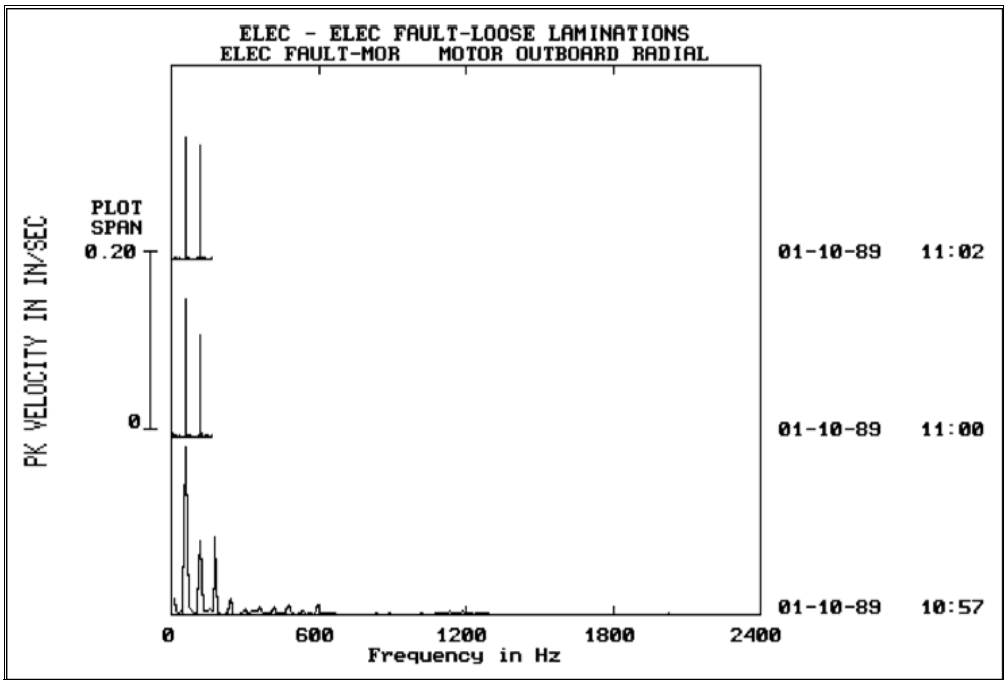


Figure 9

Date	Time Stamp	Resolution
01-10-89	10:57	400
01-10-89	11:00	800
01-10-89	11:02	3200

Table 2

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

When all the spectra in Figure 10 are expanded in an effort to separate discrete peaks, the lesser resolution data now appears as broad based peaks around 1xTS and 2xTS, while the higher resolution data tends to separate the peaks better. The data is virtually the same. However, notice that the 3200 line spectrum implies that those broad based peaks are actually the product of several peaks contained in the same cell.

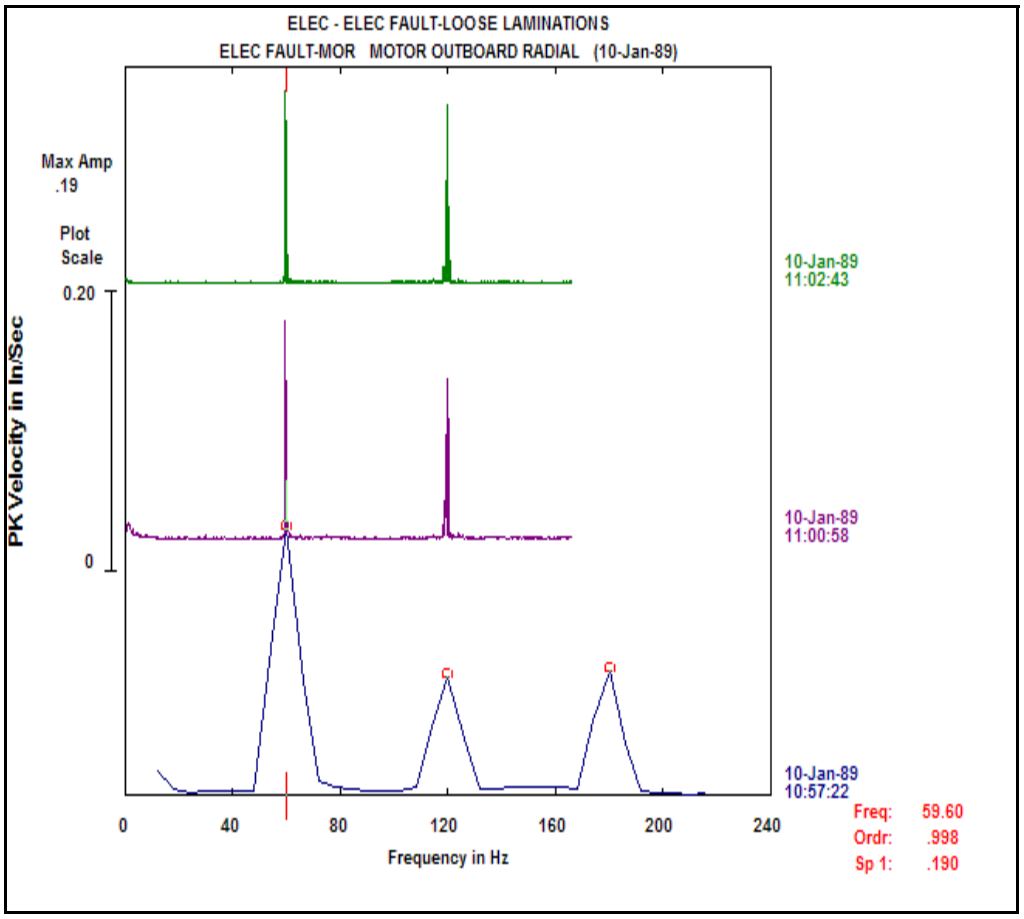


Figure 10

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Figure 11 is the single spectrum with 3200 lines of resolution. The turning speed harmonics are more discernable. It is now evident that what appeared to be a single peak at twice turning speed is really several peaks. There are several frequencies showing up around 2xTS. One peak indicates the possibility of some type of electrical defect

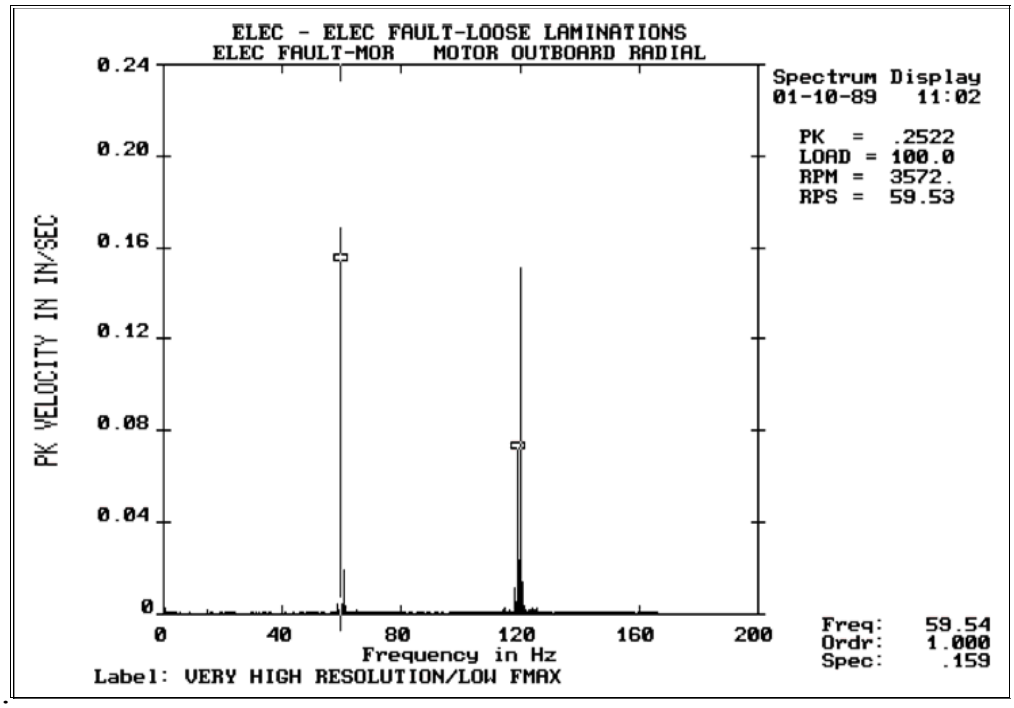


Figure 11

What other condition may be present in this equipment?

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

To help identify the specific peaks around 120 Hz, the spectrum is limited to a range between 112 Hz and 128 Hz. Examine the peaks around 120 Hz. The peak at 119 Hz was identified as 2xTS which may indicate a possible misalignment condition. Excessive misalignment could cause enough disturbances to the electromagnetic field to generate an elevated $2F_L$ peak. (See Figure 12)

$$Slip\ Frequency\ =\ Magnetic\ field\ speed - run\ speed$$

$$Slip\ Frequency\ =\ 60Hz - 59.54Hz$$

$$Slip\ Frequency\ =\ 0.46Hz(\ 27.6\ RPM)$$

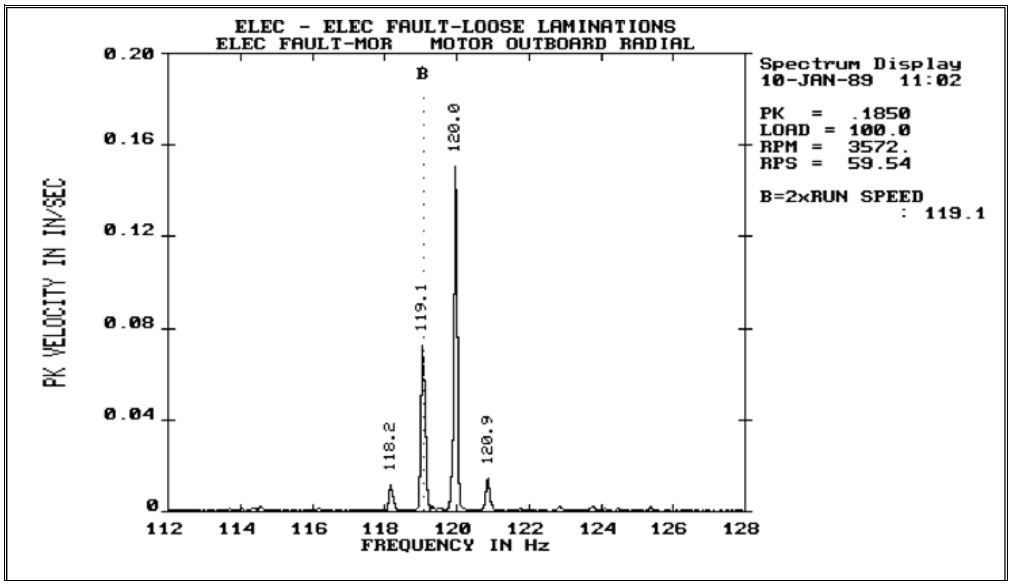


Figure 12

The sidebands around the 1xTS peak should be spaced at a frequency equal to the number of poles on the motor times the slip frequency.

$$Sideband\ =\ Number\ of\ poles \times slip\ frequency$$

$$Sideband\ =\ 2 \times 0.46Hz$$

$$Sideband\ =\ 0.92Hz$$

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The peaks at 110.2, 119.1, and 120.9 are good candidates for Poles x Slip Frequency sidebands which would suggest an electrical anomaly. The spacing between them is approximately 0.91 Hz. This is equal to the poles x slip frequency for this 2 pole motor. (See Figure 13)

The sideband spacing of 0.9 Hz around 1xTS of the motor approximates this frequency. Finding these sidebands might cause you to suspect a defective rotor bar.

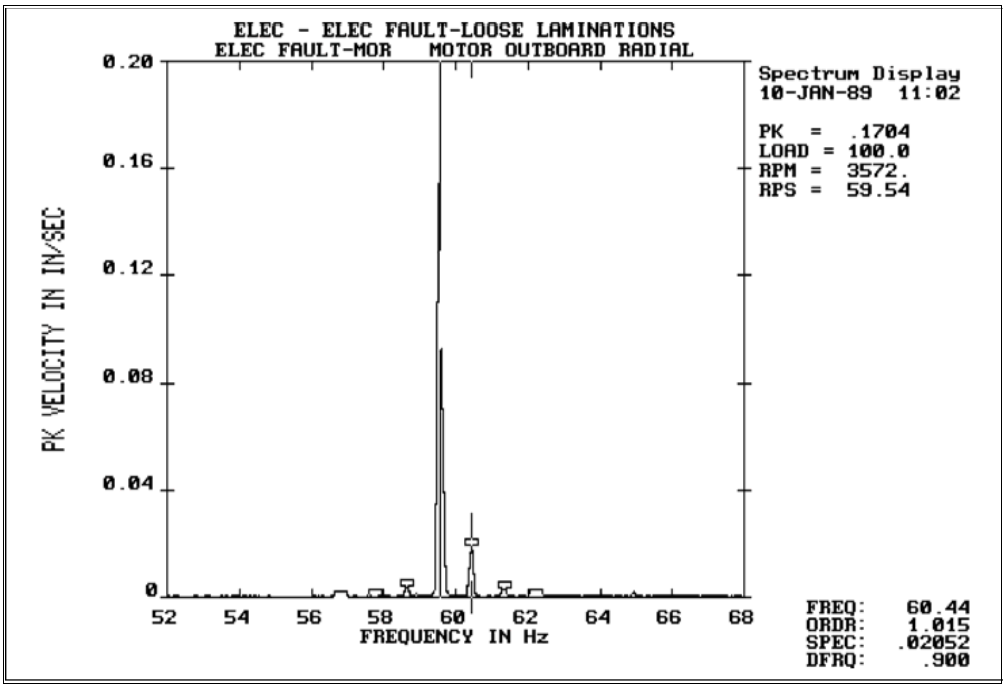


Figure 13

When the sidebands caused by the possible rotor defect are less than 20 dB down from the actual run speed peak, then a significant problem may exist.

Note

In general, amplitudes of 0.05 in/sec and higher at 120 Hz are a cause for concern and should be investigated further using some off line testing technologies.

Note

To accurately diagnose electrical defects, capturing quality resolution data is imperative. A common set up would be an Fmax of 400 Hz and 400 LOR. Collect electrical nature data in the horizontal direction unless the rotor is running off magnetic center.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Rotor bar defects will tend to generate elevated $2F_L$ and/or turning speed peaks with poles x slip sidebands as seen in Figures 14 and 15. To help determine the severity of this condition view the data in velocity dB instead of peak amplitude. When the "dB down" from each respective defect frequency (turning speed and $2F_L$) to the sideband is less than 20dB, there is cause for concern. Additional rotor testing may be in order.

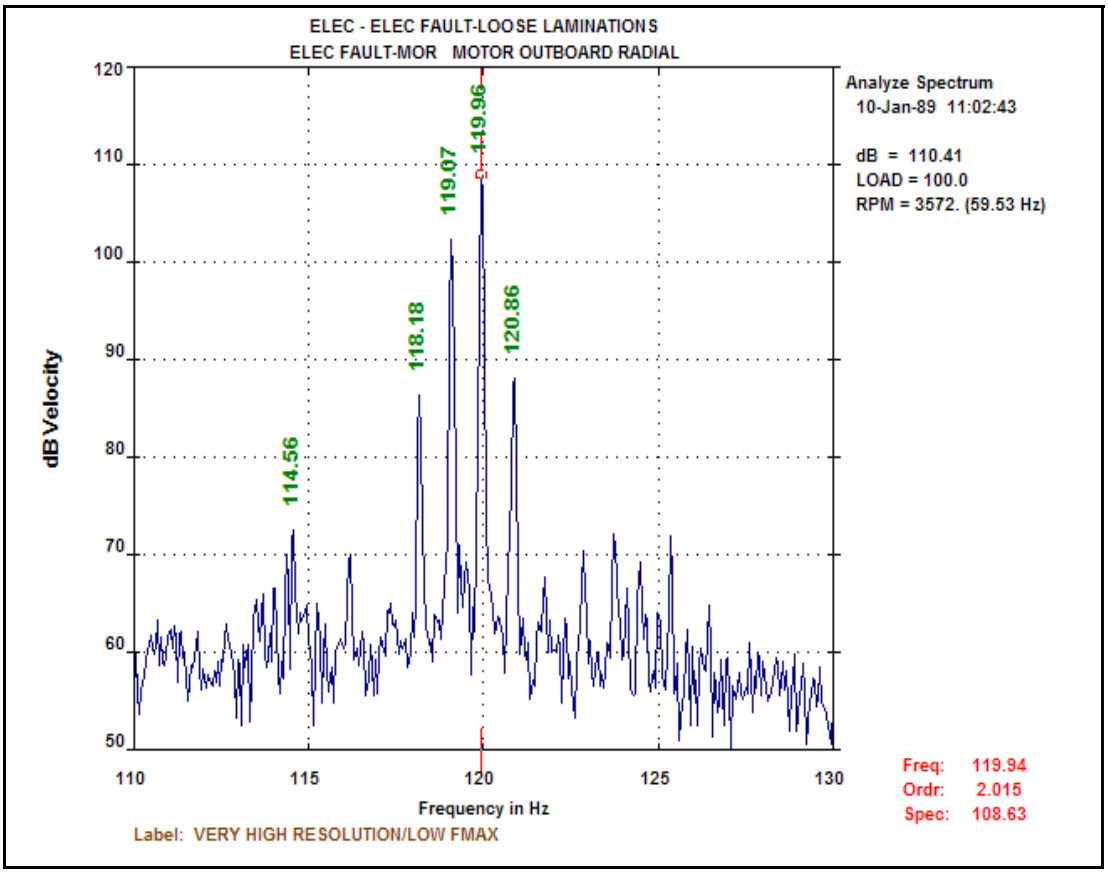


Figure 14

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

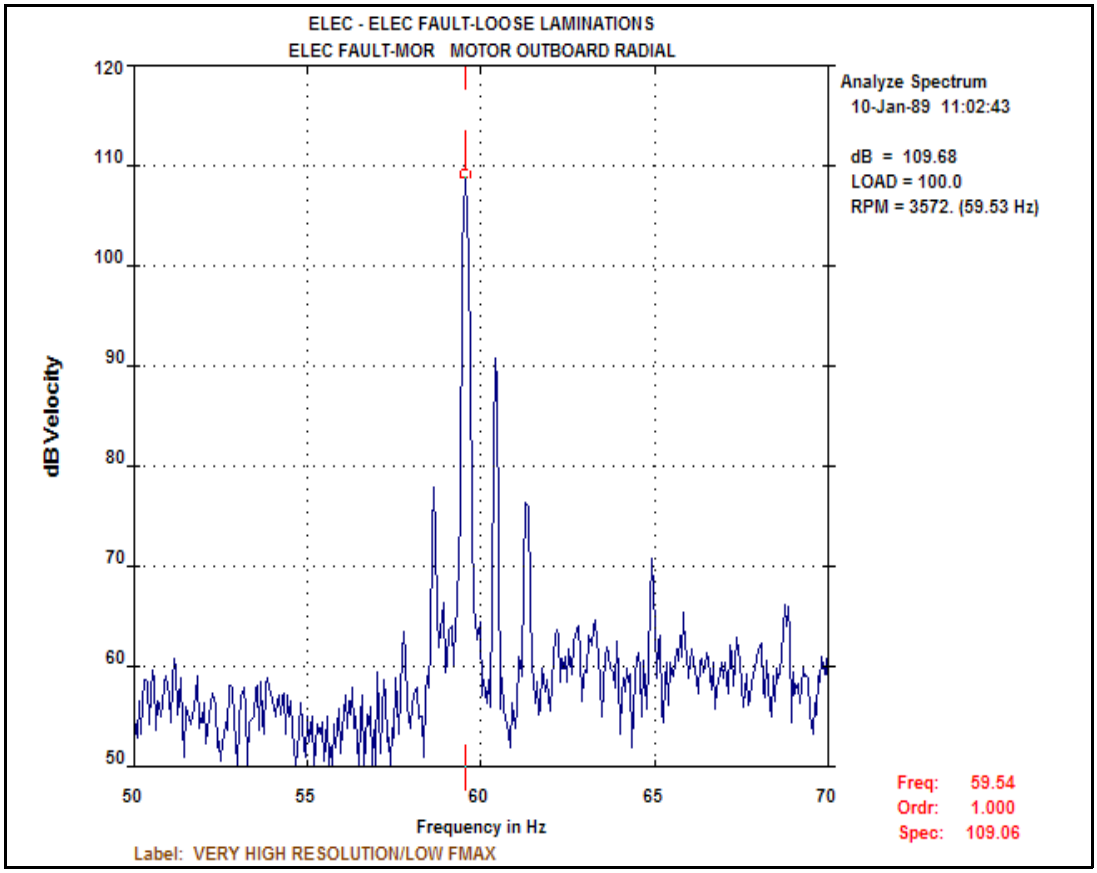


Figure 15

Data Analysis:

- Elevated (greater than 0.05 "/sec.) $2F_L$ peak with poles x slip side-bands.
- Significant TS peak with poles x slip sidebands.
- Sidebands less than 20 dB down in spectrum viewed in dB.
- 2xTS approached 50% of 1xTS peak.

Diagnosis:

- Possible defective stator; loose laminations
- Rotor integrity is suspect
- Shaft alignment is questionable

Corrective Action(s):

- Thoroughly inspect rotor for defective bar(s) or cracked end rings
- Have stator tested for lamination defects
- Align shafts during commissioning

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #2 - Boiler Feed Pump Electrical Defect

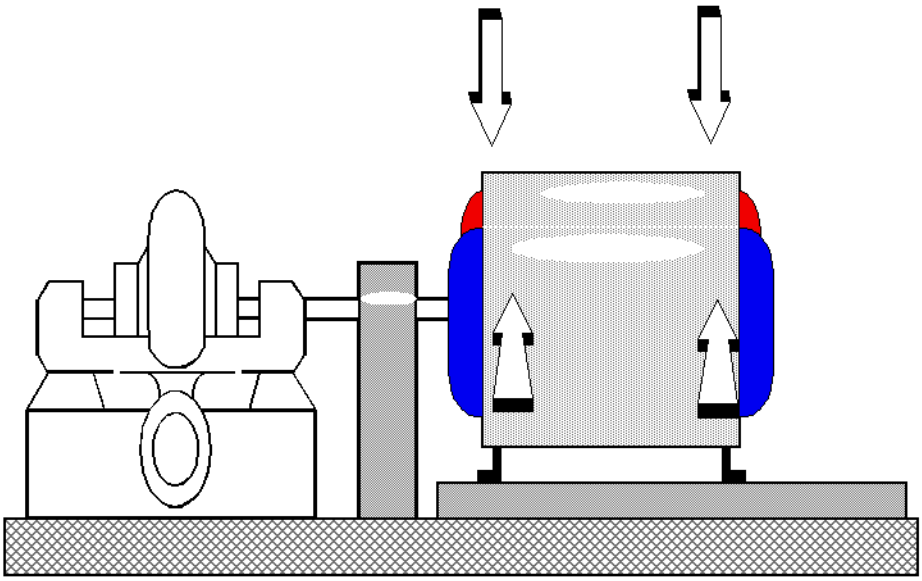


Figure 16

Equipment Information	
Horsepower	300
Shaft Speed	59.66
Number of Poles	2
Synchronous Speed	60 Hz
Type Motor	Induction

Additional Information:

- Motor supply breaker has a history of tripping intermittently.
- Significant speed fluctuations were documented.
- Current surges of 47 amps were measured.
- Slip Frequency = 0.34 Hz
- Poles x Slip Frequency = 0.68 Hz
- High resolution is essential.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

The spectrum in Figure 17 was taken on a boiler feed pump suspected of having some type of electrical defect. Vibration data was collected after the stator was tested and determined to be in satisfactory condition. Turning speed is located at 59.66 Hz. The broad skirt on the peak is suspicious. Further analysis is necessary.

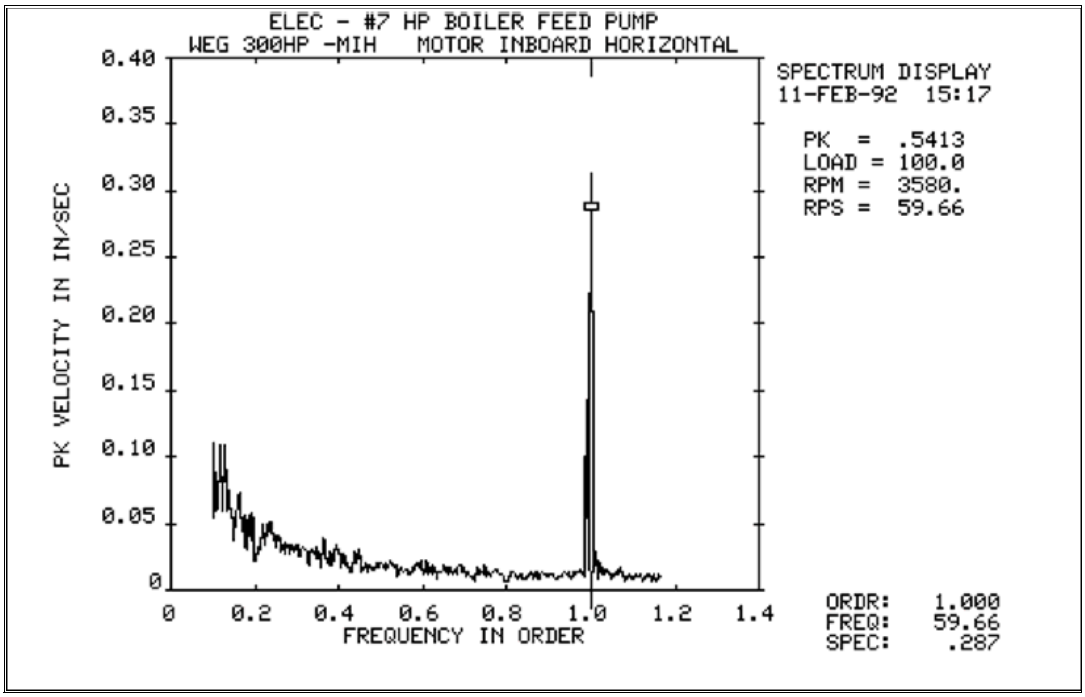


Figure 17

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Figure 18 illustrates an expanded spectrum from the motor inboard horizontal data point. What appeared to be a broad skirt peak at turning speed is now proven to be turning speed with apparent sideband(s). Without adequate resolution this defect may have been diagnosed as simple unbalance.

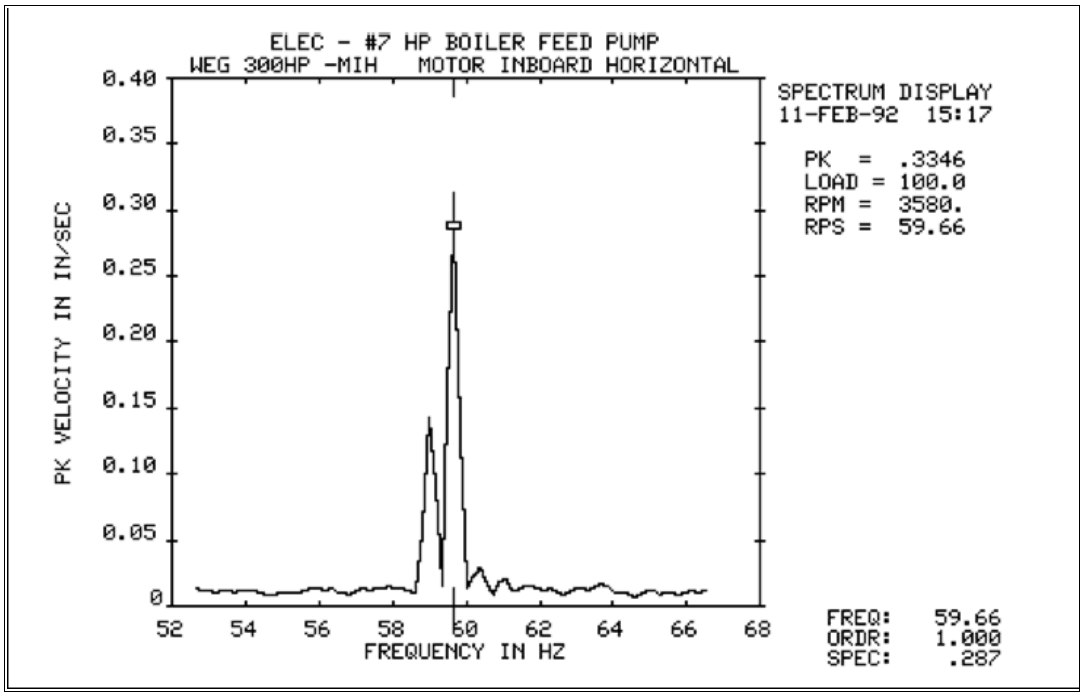


Figure 18

This demonstrates the importance of collecting high resolution data when an electrical defect is suspected. What could be the possible cause of the motor running speed sideband?

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

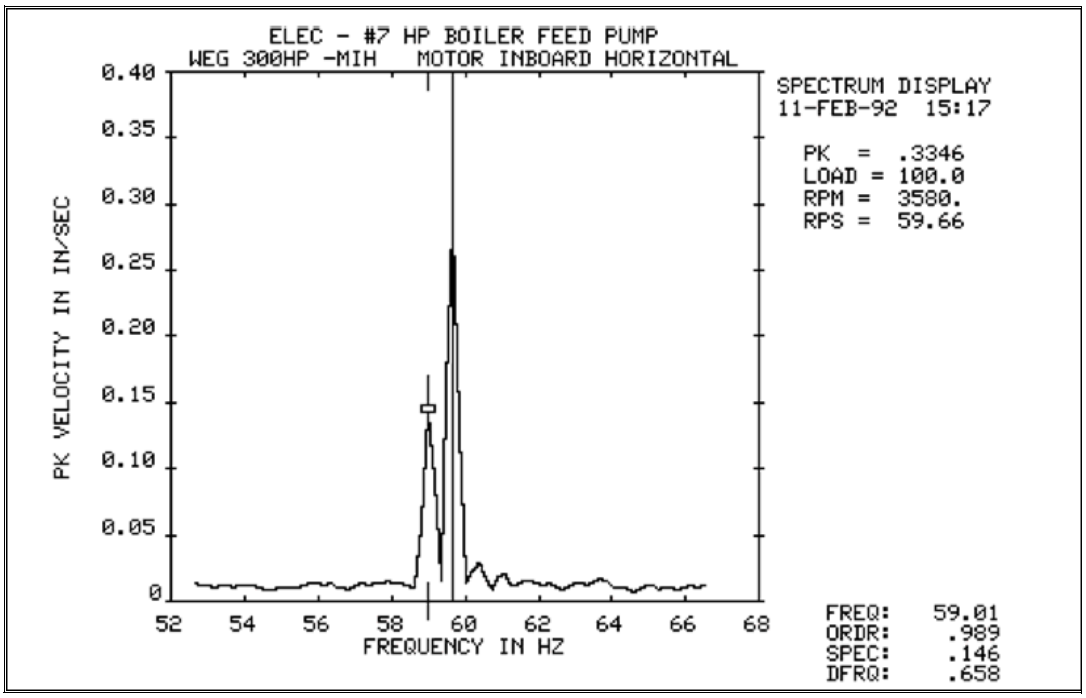


Figure 19

The difference in frequency between the turning speed peak and the lower sideband is determined by setting turning speed as the reference then moving the cursor to that sideband and reading the difference in frequency (DFRQ) in the plot. In Figure 19 that difference is 0.66 Hz, which approximates the poles x slip frequency of the motor.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

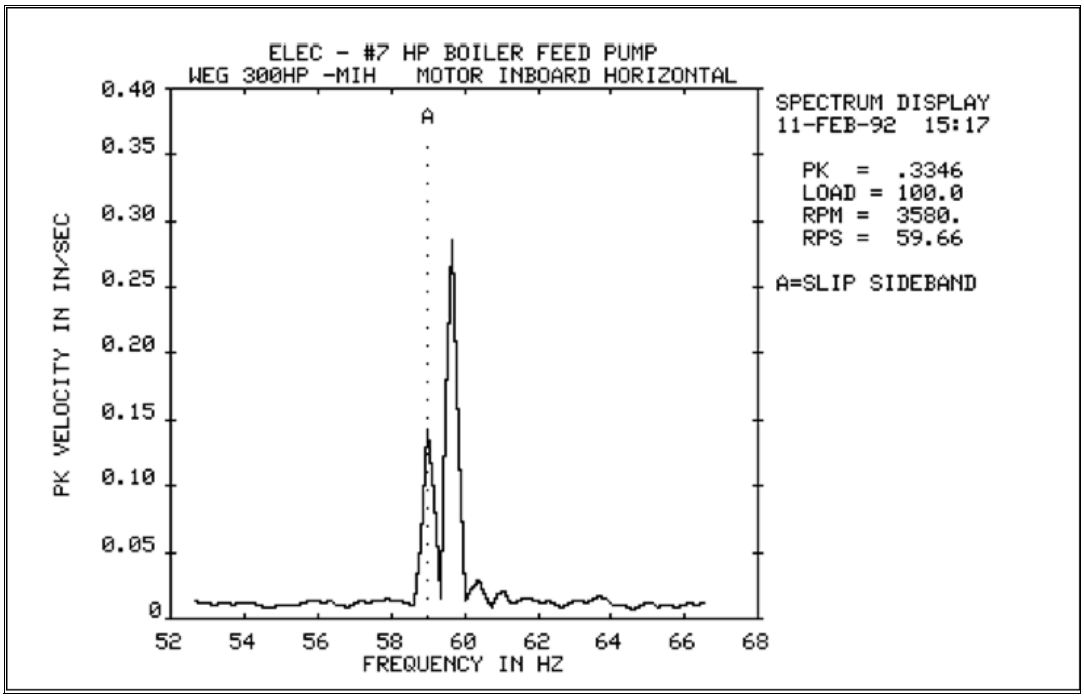


Figure 20

The presence of poles x slip frequency sidebands are good indications of rotor bar problems. (See Figure 20.) Current measurement analysis may help verify a rotor bar defect. The motor shop inspected this rotor and discovered five (5) open bars.

Had unbalance been the sole diagnosis of this machine the real problem may have gone undetected. Frequently when this happens the motor is disassembled and the rotor is "balanced" without addressing the root cause. The "repaired" motor may have been put in storage then recommissioned at some later date, only to have the original problem appear again.

Note

As with all newly installed equipment it is very important to take readings on electric motors when they are first started up and then again when the machine has come up to temperature.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Case History #3 - Kiln Drive Motor - Electrical Defect

This motor drives a large gear-box that drives a gas fired Kiln Dryer 12 feet diameter X 75 feet long.

Notice the harmonics of line frequency (60 Hz) in Figure 21. Note the amplitude at 60 Hz is 0.06 in/sec. This was reason for concern and was reported to maintenance.

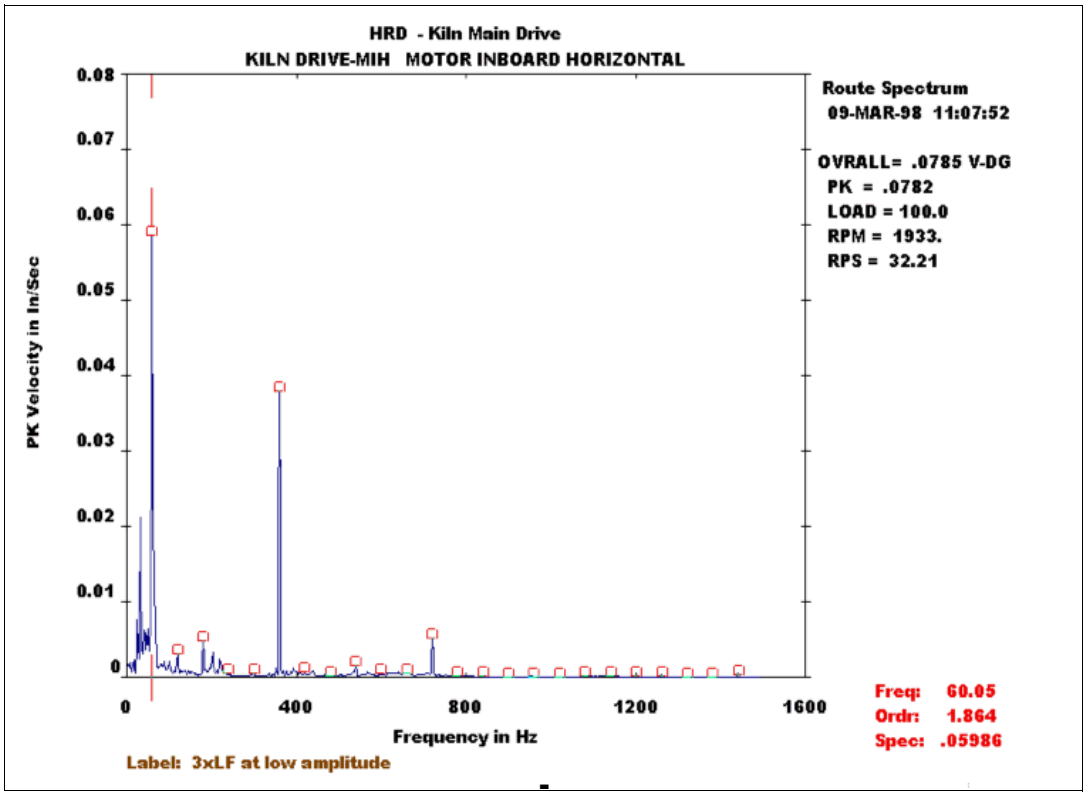


Figure 21

This motor ran for approximately 6 months without any corrections being made before finally losing an SCR. This problem was attributed to mismatched SCR cards and poor quality cards.

There will be times when you see line frequency or 2 x Line frequency and it is not a serious situation. You may have to watch it for a while to determine if you need to report it.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Vibration Problems in Electrical Systems

Most vibration problems associated with electrical systems are motor related. However, you should not overlook other sources for the vibration you may encounter.

For example, something as simple as the dress of the conductors in the raceway can cause vibration. Other causes include loose laminations in power transformers, SCR pulses in speed control systems, unbalanced phase currents, and high current pulses from welders or solenoids.

Electrical discharges can also occur in motors and generators. These discharges usually fall into one of the following categories:

- Partial discharge within the stator bar insulation
- Slot discharge between the stator bar insulation and the stator core
- Surface discharge over the end winding
- Discharge between broken conductors

Because these discharges often generate very high frequencies, you cannot detect them in frequency domain spectral analysis. Depending on the fault, you may sometimes see the discharge in the time domain. They are best detected using a high frequency oscilloscope.

The mechanical forcing functions already discussed in this class also occur in electric motors. These forcing functions include:

- Unbalance
- Thermal bow in the rotor
- Shaft or stator resonances
- Misalignment – both mechanical and electrical
- Defective bearings
- Looseness
- Rubs

Rotor bars rank second only to bearings as the main cause of motor failure. You have to detect rotor bar faults at an early stage of development. When the motor starts, especially under load and across the line, high currents flow in the rotor bars. This flow causes high rotor bar stress. Rotor problems are inevitable after numerous starts take place.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Common Induction Motor Failure Mode:

1. A rotor bar cracks because of repeated high-current stress.
2. Spot heating occurs at the crack, which may cause rotor bow. This bow looks like unbalance in a vibration spectrum, so you might balance the motor again instead of analyzing for rotor faults.
3. The bar breaks and arcing occurs, which causes additional heating and rotor bow. Even though the motor is balanced again, the rotor may rub the stator.
4. The adjacent bars carry more current, which subjects them to even higher thermal and mechanical stresses.
5. Rotor laminations are damaged, which leads to motor failure.

One of the goals that should be set for a good Predictive Maintenance program should be the development of a ***Motor Repair/Specification Sheet***.

There is no reason why an old motor can not be rebuilt to run as smoothly as a new one. As a suggestion, deal only with repair shops that have the ability to test your motors under load. It may cost a little more up front, but in the long run, thousands of dollars a year will be saved. Visit your repair shop when they have a motor they are repairing for you.

Ask questions; that is the way you will learn more about your motors. When a repair shop sees you getting involved with the work they are doing for you, the quality of work performed usually improves.

A Vibration Specification/Repair Sheet is offered on the next page.

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Motor Vibration Specification and Repair Sheet

The following information must be filled out on all new or rebuilt motors.

Date _____ **Purchase Order No.** _____
Manufacturer _____ **Company No.** _____
Frame Size _____ **Serial No.** _____
Type _____ **Model No.** _____
Class _____ **Horse Power** _____

Rated Speed _____ **Brush Part No.** _____
Full Load Amps _____ **No. of Rotor Bars** _____
Voltage _____ **No. of Stator Slots** _____
D.C. Fields (A.C.) Drop _____ **No. of Poles** _____
Hi Pot Volts (Stator) _____ **(Field)** _____ **No. of Phases** _____
Full Load Test on all Rewinds _____
Ducter Test _____ **Bar to Bar** _____
No Load Amps --- 1. _____ **2.** _____ **3.** _____ **No Load Speed** _____

(Bearings)

Coupling End Brg. (Removed) Manufacturer _____ **Brg. No.** _____
Coupling End Brg. (Installed) Manufacturer _____ **Brg. No.** _____

Opposite End Brg. (Removed) Manufacturer _____ **Brg. No.** _____
Opposite End Brg. (Installed) Manufacturer _____ **Brg. No.** _____

(Vibration Amplitude Data)(no load)

Bearing Location	Accel. Location	Overall Level 0 - 200 Orders (.05)	1 x rpm (.04)	2xLine Freq. 120 Hz (.05)	2 - 5 Orders (.02)	6-50 Orders (.02)	51 - 120 Orders (.7 g's)	120 - 150 Orders (.7 g's)
Opp. End	Horz.	_____	_____	_____	_____	_____	_____	_____
Opp. End	Vert.	_____	_____	_____	_____	_____	_____	_____
Opp. End	Axial	_____	_____	_____	_____	_____	_____	_____
Cpl. End	Horz.	_____	_____	_____	_____	_____	_____	_____
Cpl. End	Vert.	_____	_____	_____	_____	_____	_____	_____
Cpl. End	Axial	_____	_____	_____	_____	_____	_____	_____

Note:
1. **Vibration readings should be recorded in (Velocity) / In./Sec. Peak (all Bands except 51 - 120 and 120 - 250 orders) they should be recorded in Acceleration (g's).**
2. **Only one axial reading is required (either end ok) (opposite end or coupling end).**
3. **Rotors must be balanced to ISO - 1940 (G - 2.5 Grade).**
4. **The Accelerometer designated for use by the P/PM Department is:** _____

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Glossary

Line Frequency (F_L)

the frequency of the electrical supply line

Magnetic Field Frequency

the rotational frequency of the electromagnetic field

Rotor

the rotating element of the motor

Pole(s)

the magnetic locations set up inside an electric motor by the placement and connection of the windings

Rotor Bar

the ferrous bar in an induction motor rotor into which the current is induced to force the rotor to turn

Rotor Bar Pass Frequency

calculated as the number of rotor bars multiplied by the turning speed of the rotor

Rotor Frequency

the rotational frequency of the rotating element in the motor

SCR

Silicone Controlled Rectifier

Stator

the stationary element of the motor

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Stator Slot

the cavity into which the winding of an induction motor is laid

Slip Frequency

the difference between the rotational frequency of the electromagnetic field of the motor and the rotational frequency of the rotor

Slot Pass Frequency

calculated as the number of stator slots multiplied by the turning speed of the rotor

Two Times Line Frequency (2x FL)

twice the line frequency

Copyright 2006, Emerson Process Management. All rights reserved. Rev 04/06

Journal Bearings

Section 9

Objectives

- Describe journal bearings / construction
- Understand how lubrication can affect vibration levels in journal bearings.
- Realize the importance of multi-position data collection
- Determine the best data for different bearing types

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Journal Bearings

Excessive clearance, improper bearing load, and improper lubrication can all cause high vibration levels in journal bearings. A journal bearing with excessive clearance allows a small excitation force, such as a slight unbalance or misalignment, to cause significant vibration in the bearings. The predominant frequency of the vibration can occur at 1xTS, 2xTS, 3xTS, or even higher harmonics, depending on bearing design and application.

Collect data in the radial and axial directions. Radial readings usually provide the best information on plain bearings. Compare both vertical and horizontal readings. The vertical reading usually gives the best indication of excessive clearances in a journal bearing. Axial readings are best for thrust bearings.

Sleeve bearings support a shaft on a thin film of oil rather than with metal-to-metal contact. (See Figure 1) The clearance between the shaft and the bearing is commonly 0.002" to 0.008". This clearance means some looseness exists in the system, and it is common to see some harmonics of turning speed. The amplitude of the harmonics rise as the clearance becomes larger.

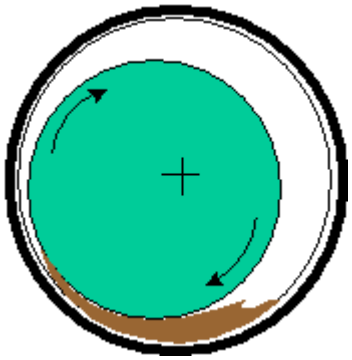


Figure 1

Oil whirl is a condition of lubrication instability that allows a cyclical rotation of the oil at or near the oil wedge created by the rotation of the shaft in the bearing. Allowed to progress, this condition may lead to oil whip which can be an extremely destructive force.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Oil whip occurs when the oil film in pressure-lubricated, sleeve-type bearings exerts a force that pushes the shaft around within the bearing. During spectral analysis, oil whip may be detected at less than one-half the shaft speed. Under normal operating conditions, the shaft rides up the side of the bearing on the oil film wedge. Due to friction, the oil film speed approximates only 42 percent to 47 percent of the shaft speed. However, the oil film force usually remains very small when compared to normal forces in the machine.

Oil whip can often by corrected by either properly loading the bearing or by changing one or more of the following: bearing design, oil viscosity, oil pressure, or the oil injection point.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Several types of sleeve bearing designs minimize the effect of oil whip. These bearings include the tilted pad bearing (Figure 2), axial grooved bearing (Figure 3), and lobed bearing (Figure 4). These bearings have surfaces that form multiple oil film wedges in an attempt to center the shaft within the bearing.

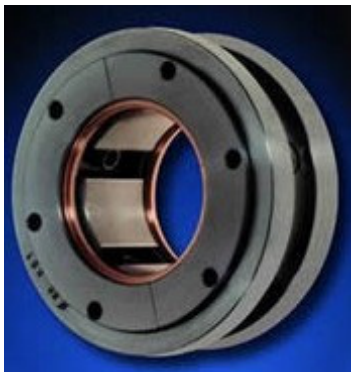


Figure 2

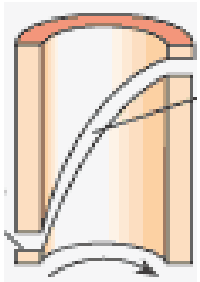


Figure 3

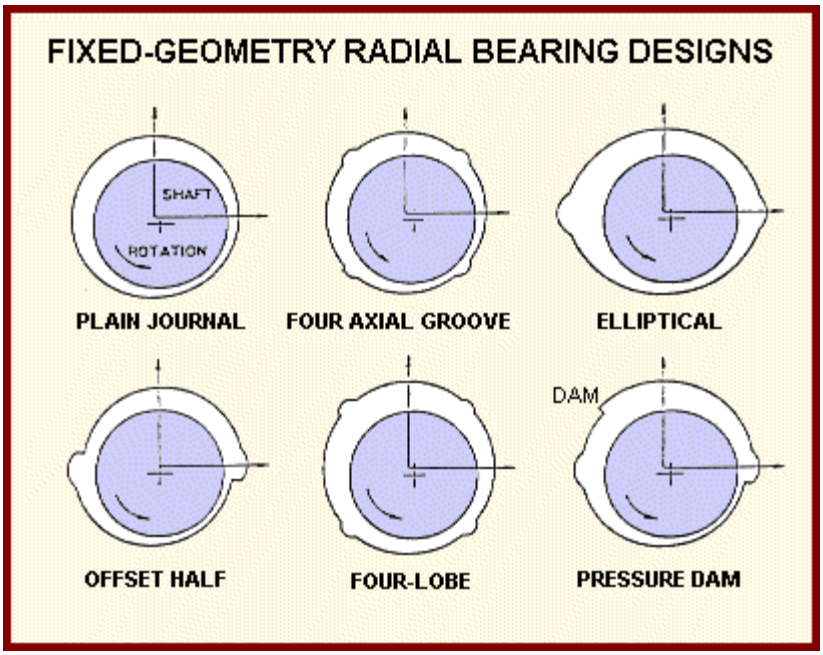


Figure 4

If the shaft is perfectly centered in a bearing, the machine is less susceptible to oil whip. The shaft can become eccentric within the bearing because of improper bearing design, improper loading, or excessive bearing wear. The oil film force can then become the dominant force within the machine.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The plot in Figure 5 shows nine harmonics of turning speed, and four of them are high. The peak just below 1xTS comes from the output shaft of the fluid drive unit. The overall value of this spectra approximates 0.15 IPS. This machine could be classified as running acceptably.

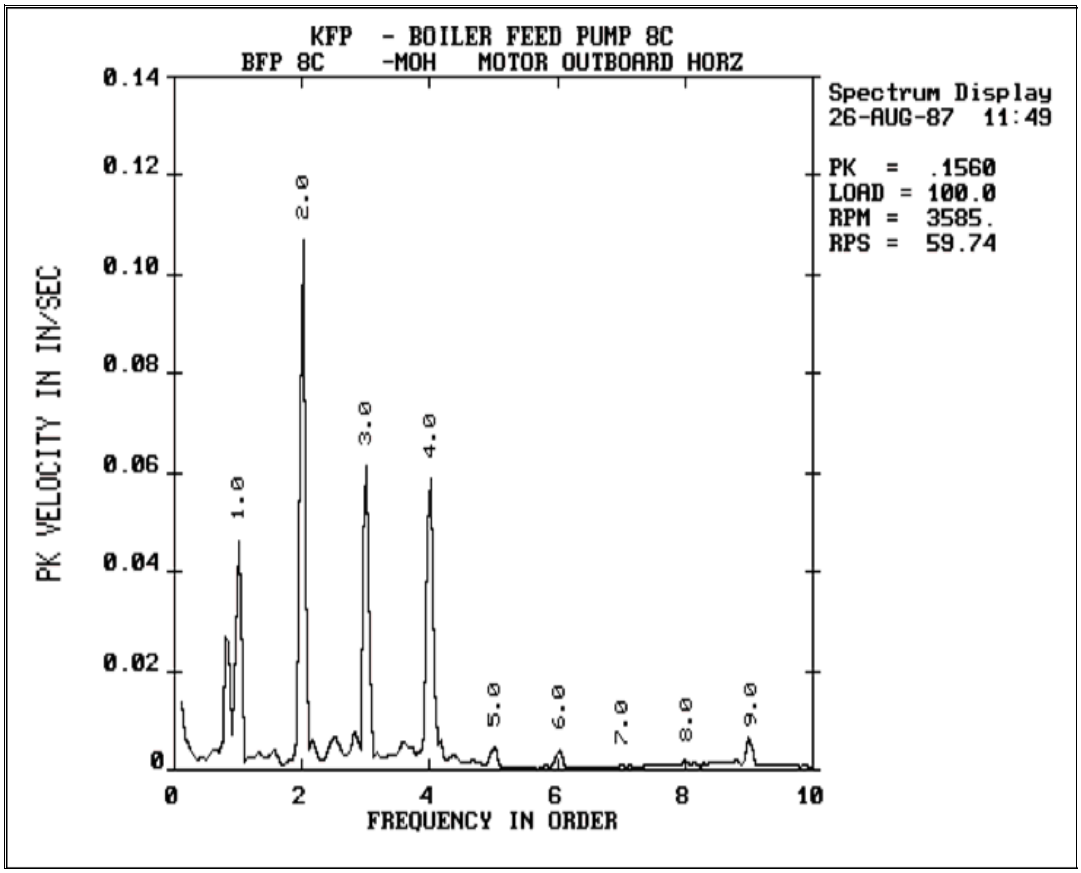


Figure 5

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Case History #1 - Direct Drive Centerhung Centrifugal Fan

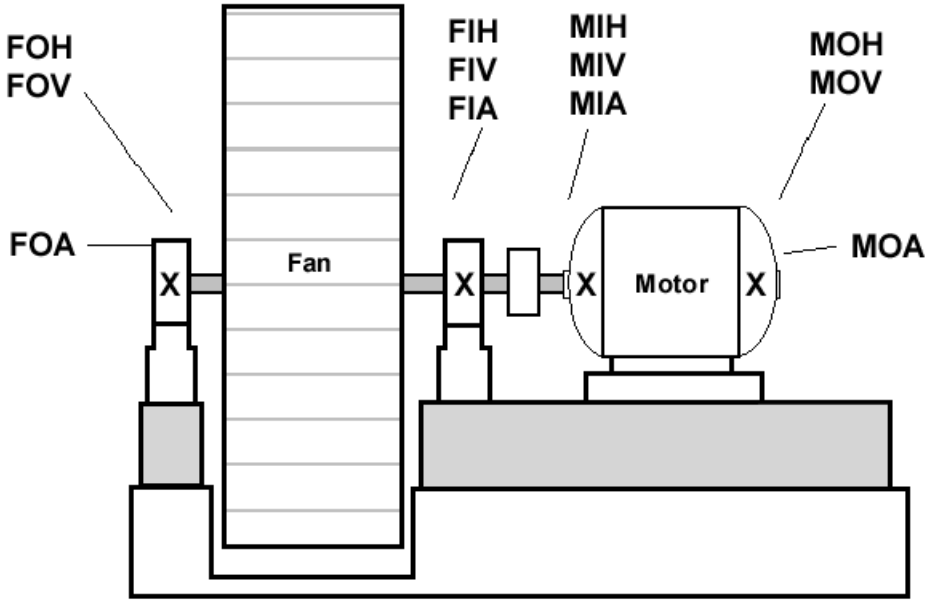


Figure 6

Equipment Information:

◆ Sleeve bearing motor, 1250 HP, 10 poles
◆ The direct driven fan uses sleeve bearings designed with thrust faces to position the fan rotor.
◆ A dynamically balanced load is placed on the fan by allowing air to be drawn in from both sides. Typically the fan will ride against the thrust face of one of the bearings.
◆ The motor and inboard fan bearing are mounted on a large concrete foundation. The outboard fan bearing has a much smaller base with a support structure made of steel.
◆ Circulating water is used to cool the fan bearings.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Data taken on the five points of this fan show very little 1xTS vibration in the radial directions. The axial vibration peak at 1xTS is the dominant peak in the data in Figure 7. The absence of significant radial vibration indicates that unbalance is not a problem. Rather, the problem most likely involves axial looseness or misalignment. Examine the motor points to check for an alignment problem.

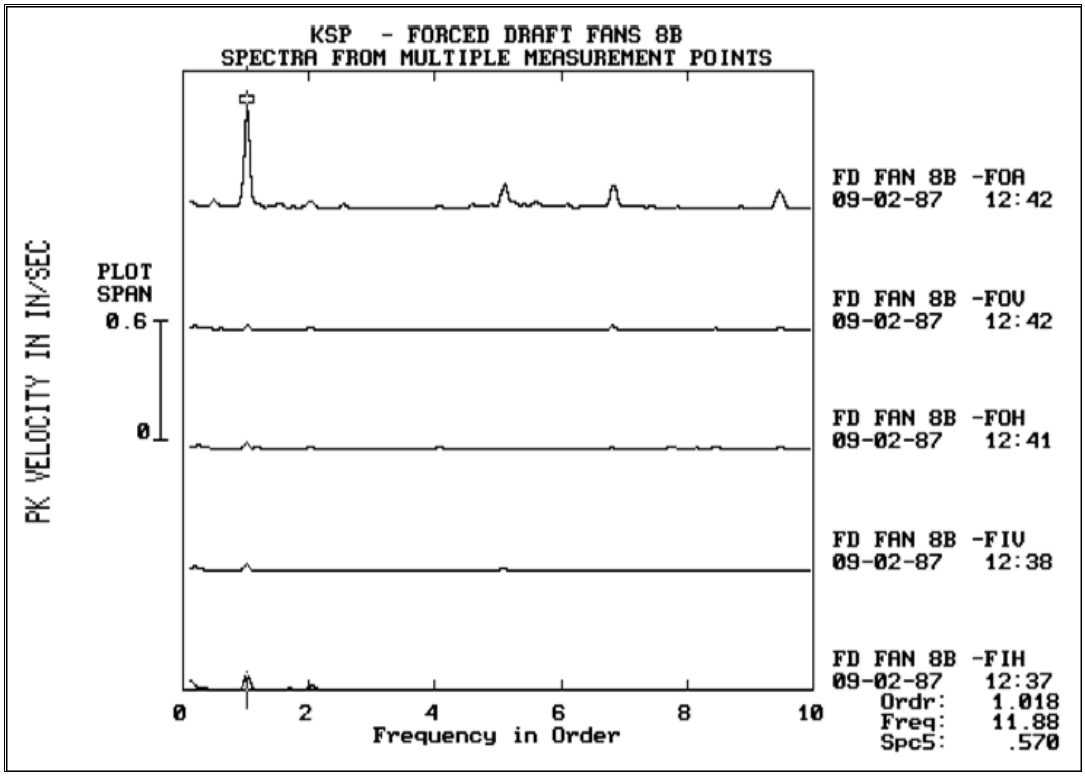


Figure 7

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Notice the relatively low scale on the motor points. The motor generates very little energy. The amplitude of the 1xTS peak in the axial direction of the fan is over 30 times higher than the 1xTS peak in the axial direction of the motor. The difference in amplitudes indicates that this problem does not involve alignment. An alignment problem normally shows as much or more vibration on the motor, because its lighter weight offers less resistance to movement by alignment forces. Therefore, the evidence points to the fan. The motor appears to be operating at acceptable levels. (See Figure 8)

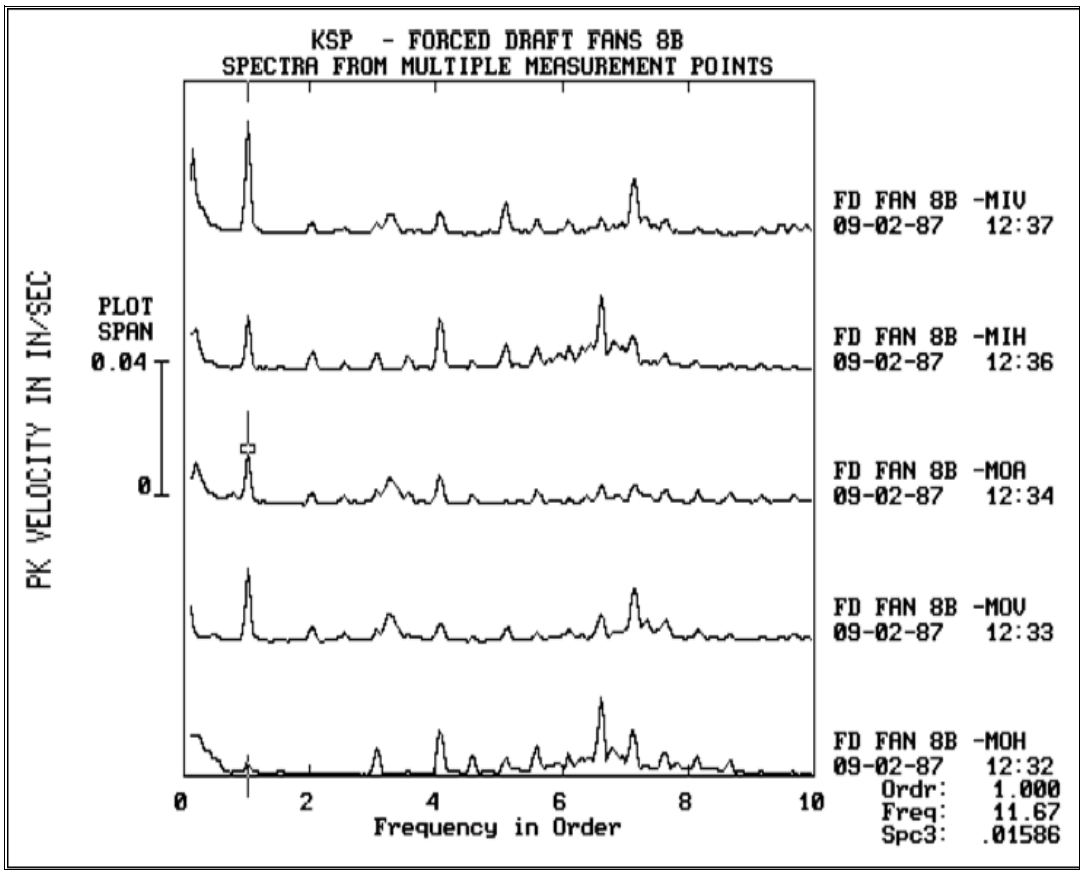


Figure 8

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The growth progression of the 1xTS axial peak of the fan Outboard Axial Point appears in the multiple spectra plot in Figure 9. Other peaks show no significant change in amplitude. Only the fan axial position exhibits high vibration. One or both of the thrust faces have possibly worn to allow axial looseness.

The radial clearances of the bearing, however, may not have changed very much with time, thereby keeping radial vibration levels low. The radial surface of the bearing has a much larger area than the axial surface, so it should wear more slowly.

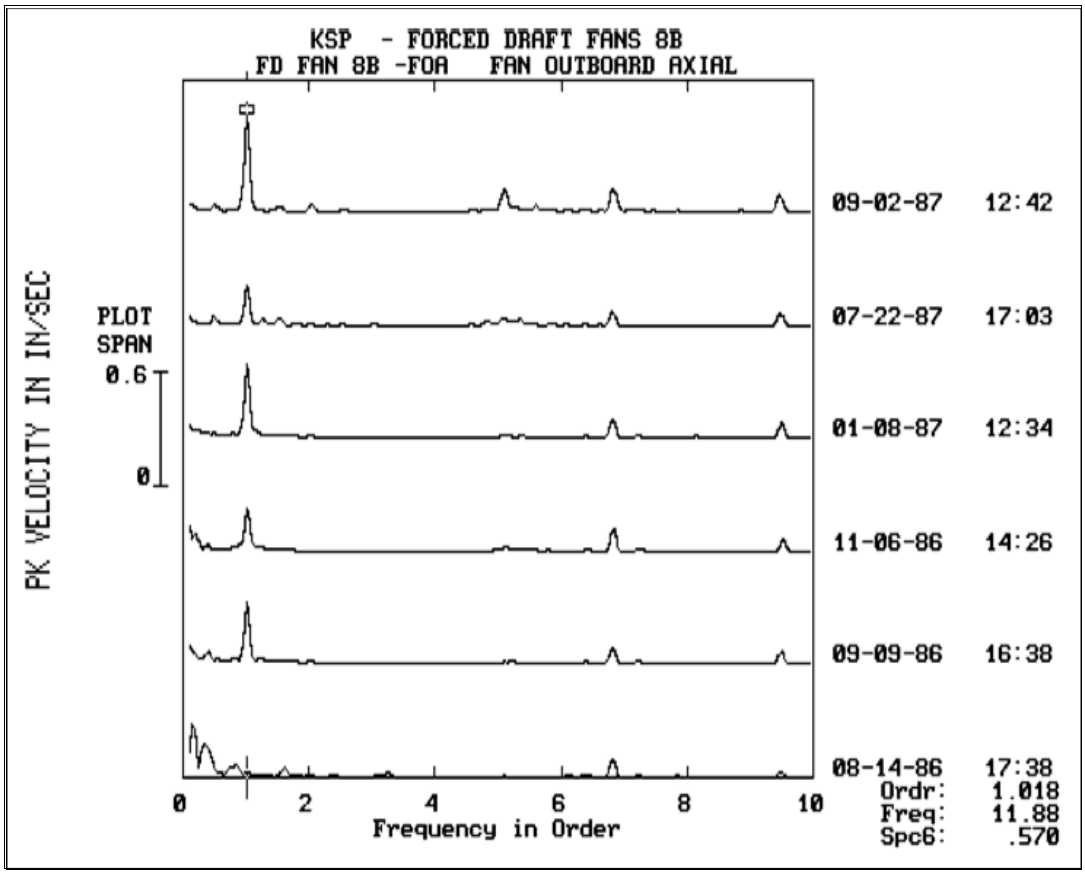


Figure 9

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

A single spectrum view of the last fan axial measurement in Figure 10 shows the dominant 1xTS peak. The other labeled peaks are apparently harmonics of 1/2 run speed. Fractional harmonics usually indicate looseness.

The thrust faces that locate this shaft have apparently worn and now allow excessive axial movement. The steel foundation under the outboard fan bearing lacks the stiffness and resistance to motion found in the concrete foundation under the inboard fan bearing. Therefore, the axial motion of the fan shaft appears more at the outboard bearing than at the inboard bearing.

This fan has a shim pack behind each of the thrust face bearings. If the babbitt on the bearings is still in good condition, properly shimming the bearings will remove the excess axial play.

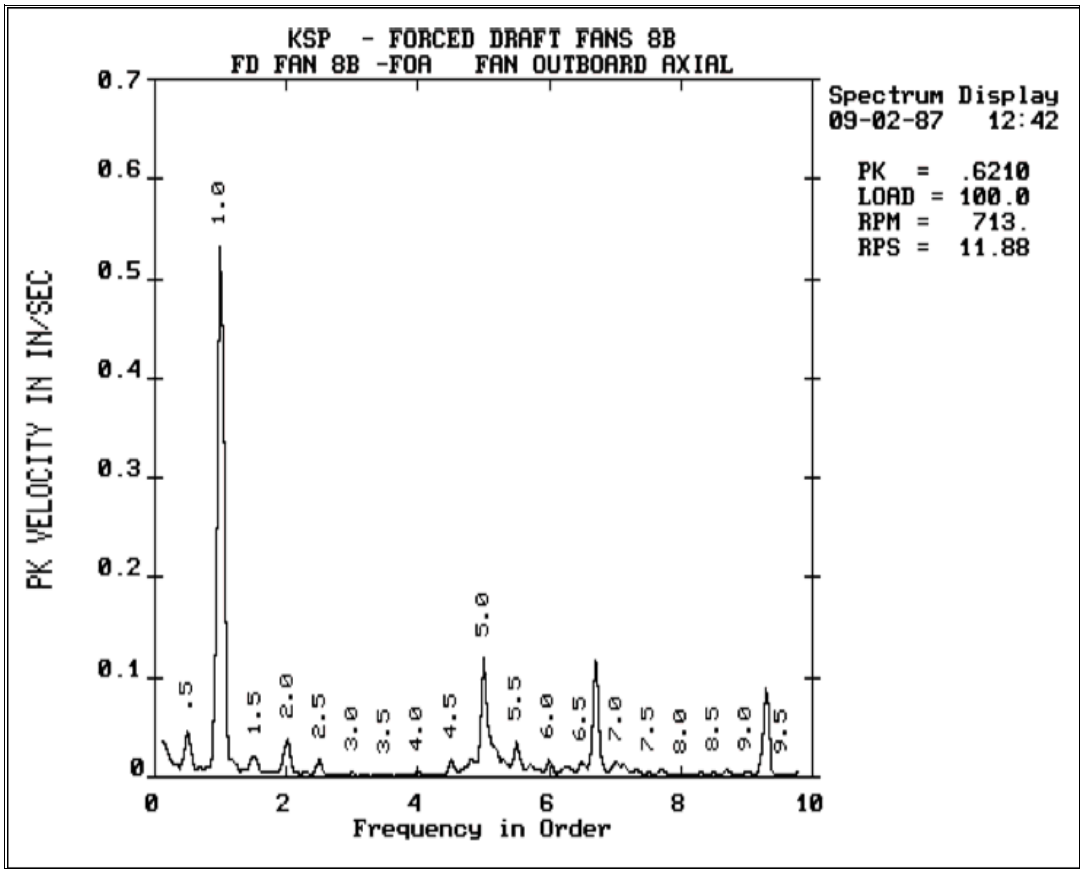


Figure 10

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

A short portion of the time waveform appears on the plot in Figure 11. The vertical lines mark the time required for the shaft to complete one revolution. Even though a 1xTS peak dominates the spectrum, the waveform is neither repeatable nor periodic. This evidence indicates looseness, because unbalance and misalignment appear more periodic.

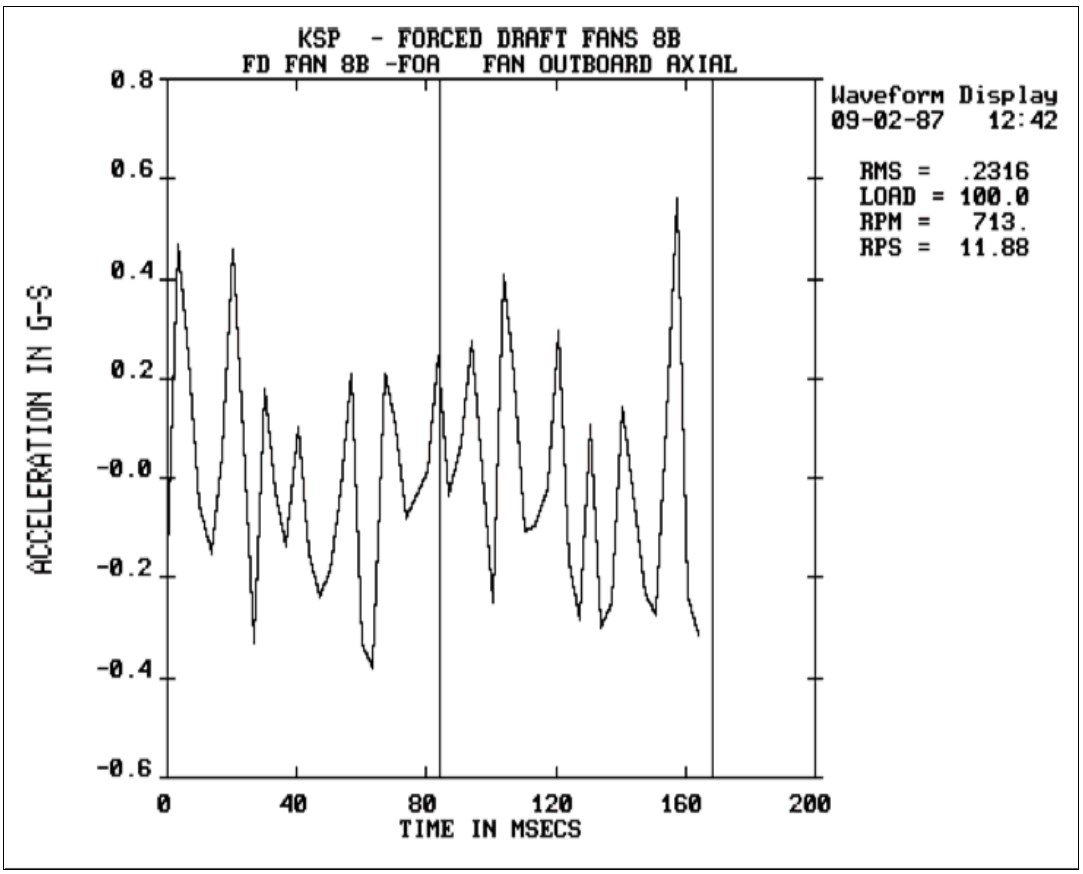


Figure 11

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Case History #2 - Turbine Generator Set

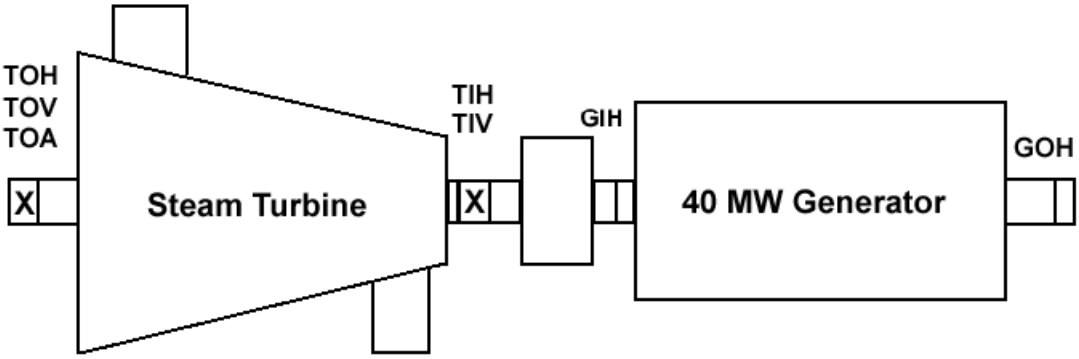


Figure 12

Equipment Information:

◆ 40-megawatt turbine generator
◆ This unit has a history of vibration problems, particularly on the turbine bearings.
◆ A common oil sump supplies lubrication to both the generator and the turbine. Therefore, both units receive oil at the same temperature and pressure.
◆ The unit is equipped with displacement probes wired into a panel. All the data shown for this unit displays in displacement.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Oil whirl peaks appear predominantly at 0.4xTS on the turbine outboard bearing positions - TOH and TOV. The highest vibration levels appear at 1xTS on the inboard bearing positions of the turbine - TIH and TIV. Note that 1xTS peaks normally have the highest amplitudes. The 0.4xTS peak also appears on the generator positions GIH and GOH. Under normal conditions a peak should appear at about 0.4xTS, but amplitudes should remain low and steady. Watch for stability by monitoring a live spectral display.(See Figure 13)

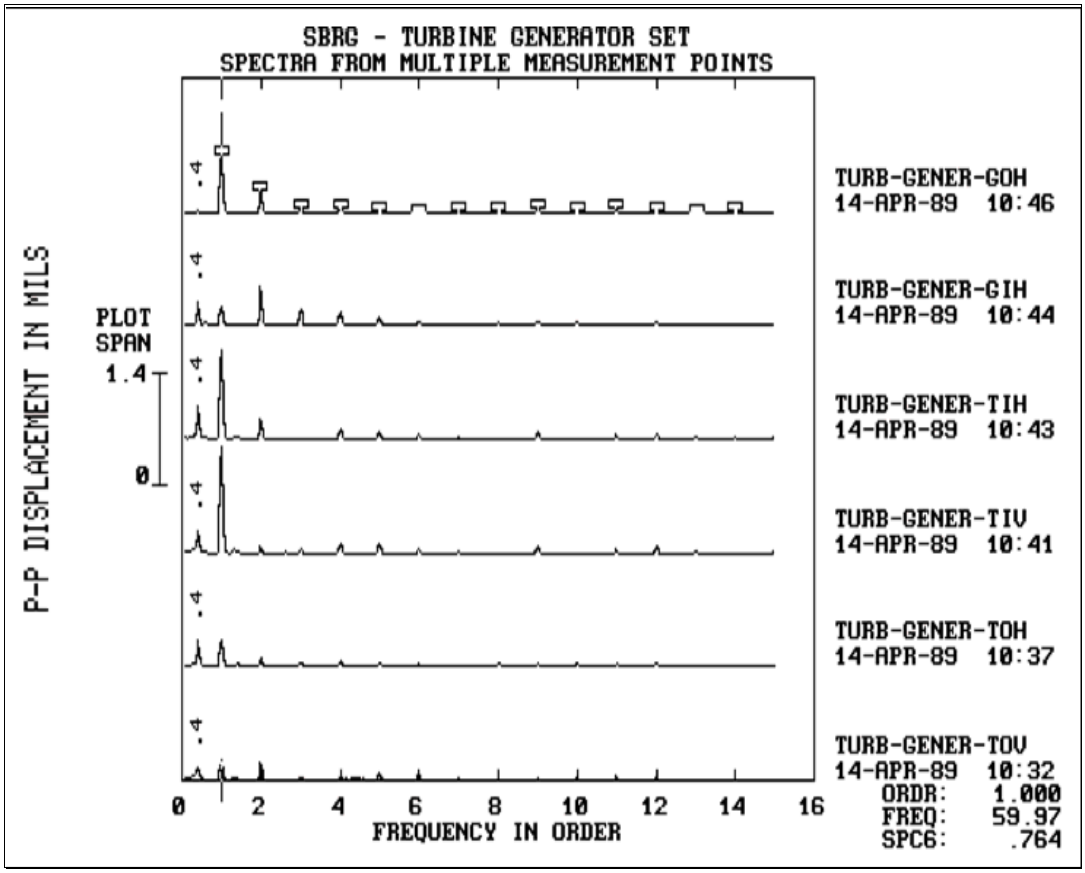


Figure 13

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The spectra in Figure 14 shows the effects of peak-hold averaging on the outboard turbine points. The spectra dated 14-Apr-89 - 10:32, and 10:37 come from the regular route mode. The second and top spectra come from the use of peak-hold averaging. Peak-hold averaging keeps the highest value measured among all the averages for each line of resolution.

This averaging mode reveals that the amplitude for the oil whirl peaks surpassed that for the 1xTS peaks. Even though the amplitude at both frequencies is relatively low, 0.4xTS peaks that exceed 1xTS peaks indicate a significant problem.

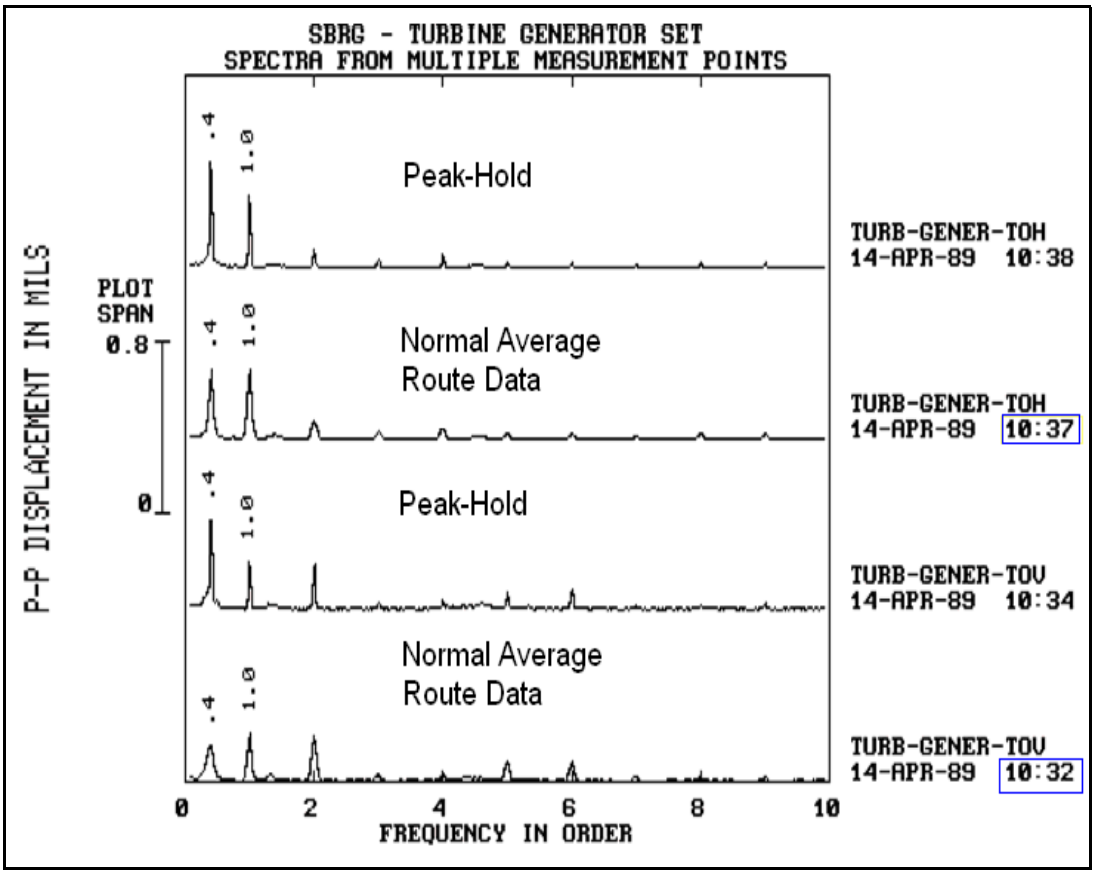


Figure 14

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The spectrum in Figure 15 shows data for the turbine outboard horizontal point - TOH. The data comes from routine route data collected with normal averaging. The height of the oil whirl peak causes concern, because its amplitude equals that of the 1xTS peak.

When viewed in live mode, this oil whirl peak appears very erratic in amplitude. It sometimes appears much higher than 1xTS, and at other times it almost disappears. An erratic amplitude characterizes an oil instability problem.

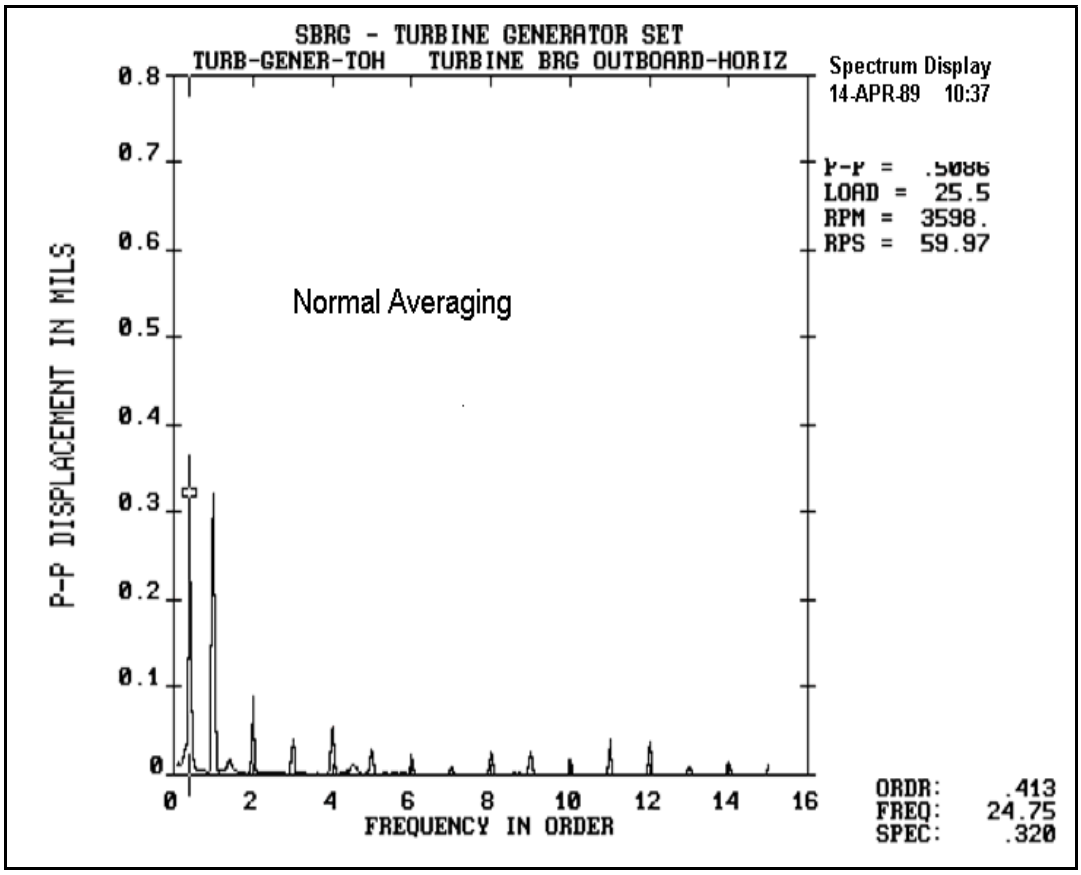


Figure 15

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The plot in Figure 16 shows the time waveform for the turbine outboard horizontal point - TOH. The vertical lines denote the time required for one revolution of the shaft to occur. It basically shows a non-repeating pattern. Every two to three revolutions of the shaft, a peak caused by the whipping motion of the oil becomes visible. The RMS value of the time waveform is 0.4613 mils peak-to-peak, but the true peak-to-peak value of the time waveform appears to exceed 0.7 mils.

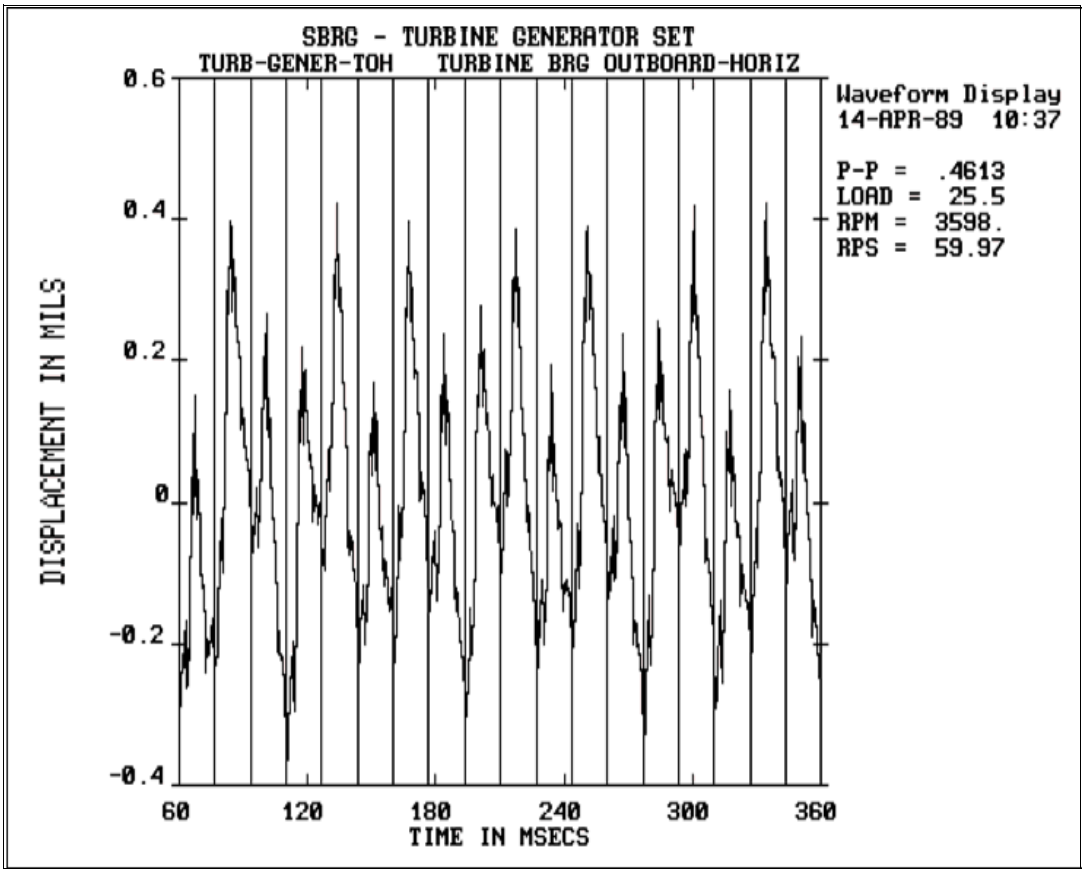


Figure 16

 Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Case History #3 - Sleeve Bearing Looseness

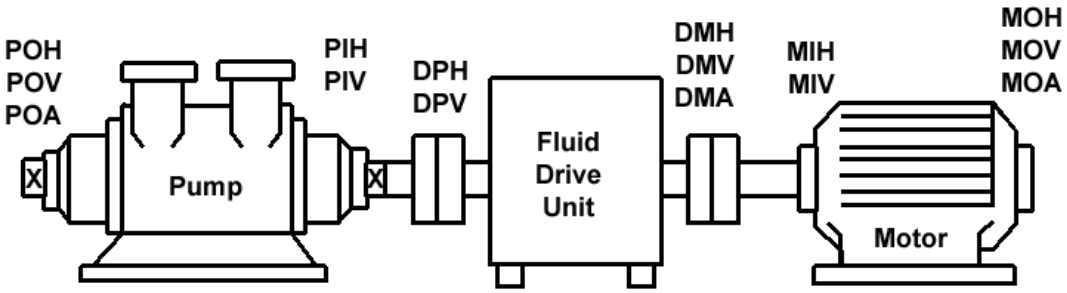


Figure 17

Equipment Information:

◆ 1250-HP, 2 pole motor
◆ The fluid drive unit has a design similar to a torque converter. It allows the pump to operate at a slightly slower speed than the motor.
◆ Three of these pumps are used for each boiler. One pump has come off line for major repairs, leaving just two available. If one of the remaining pumps fails catastrophically, the boiler would have to shut down.
◆ The pump is a centerhung unit with nine vanes on the impeller.
◆ All bearings on the motor and pump are pressure lubricated, water cooled, sleeve bearing designs.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The inboard vertical and horizontal points have amplitudes higher at 1xTS and 2xTS than the outboard points. The amplitudes are less than 0.15 IPS, so they probably do not indicate a problem other than some minor looseness on the inboard bearing. The 1xTS peaks for all points are low for a 3560 RPM motor. (See Figure 18)

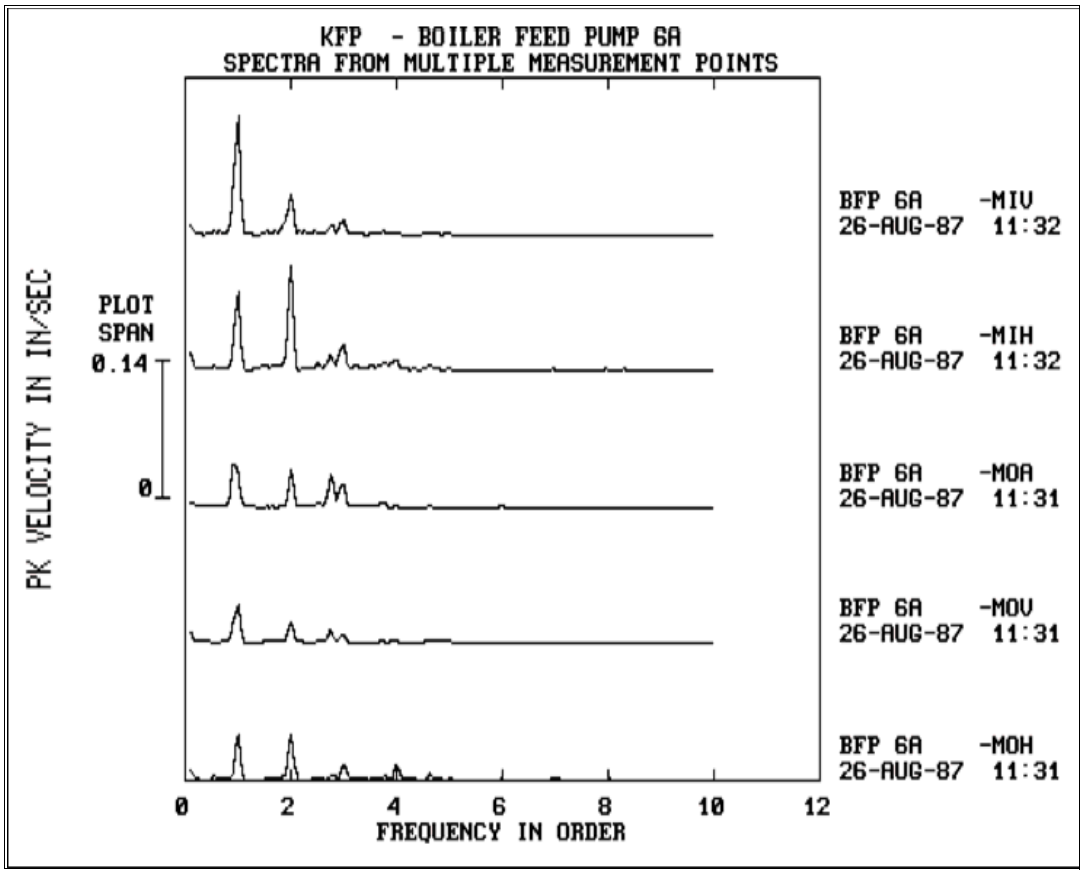


Figure 18

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The peaks on the fluid drive unit are below 0.15 IPS. (See Figure 19) Most of the other fluid drive units at this plant have higher levels of vibration. This indicates little problem in this unit. Most of the vibration occurs at 1xTS in the radial directions.

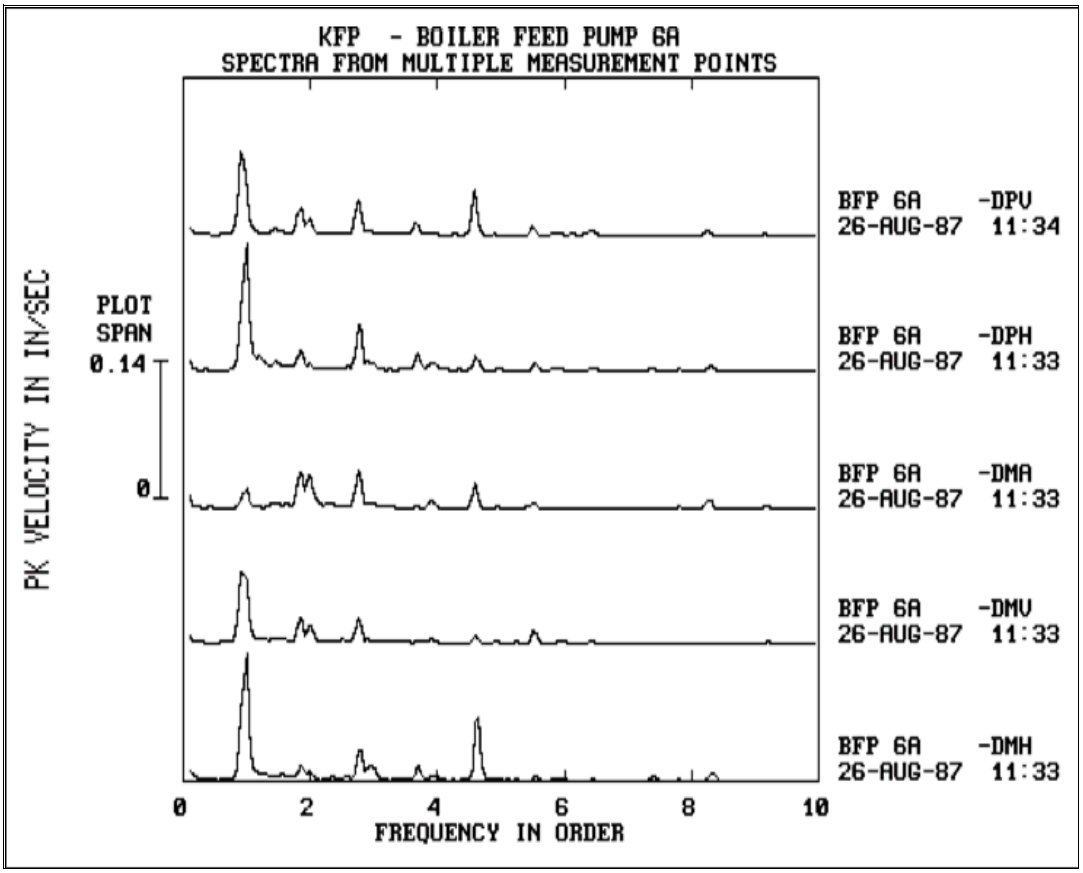


Figure 19

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The multiple spectra plot in Figure 20 shows dominant 2xTS peaks in the horizontal inboard and outboard positions on the pump. Little axial vibration exists, reducing the possibility of an alignment problem. The sleeve bearing journals have probably loosened up in the horizontal direction.

Looseness is often directional in nature. A cause of horizontal wear could be a side discharge of the pump. The pressure of the discharge flow would continuously force the shaft against one side of the bearing, causing that side to wear excessively.

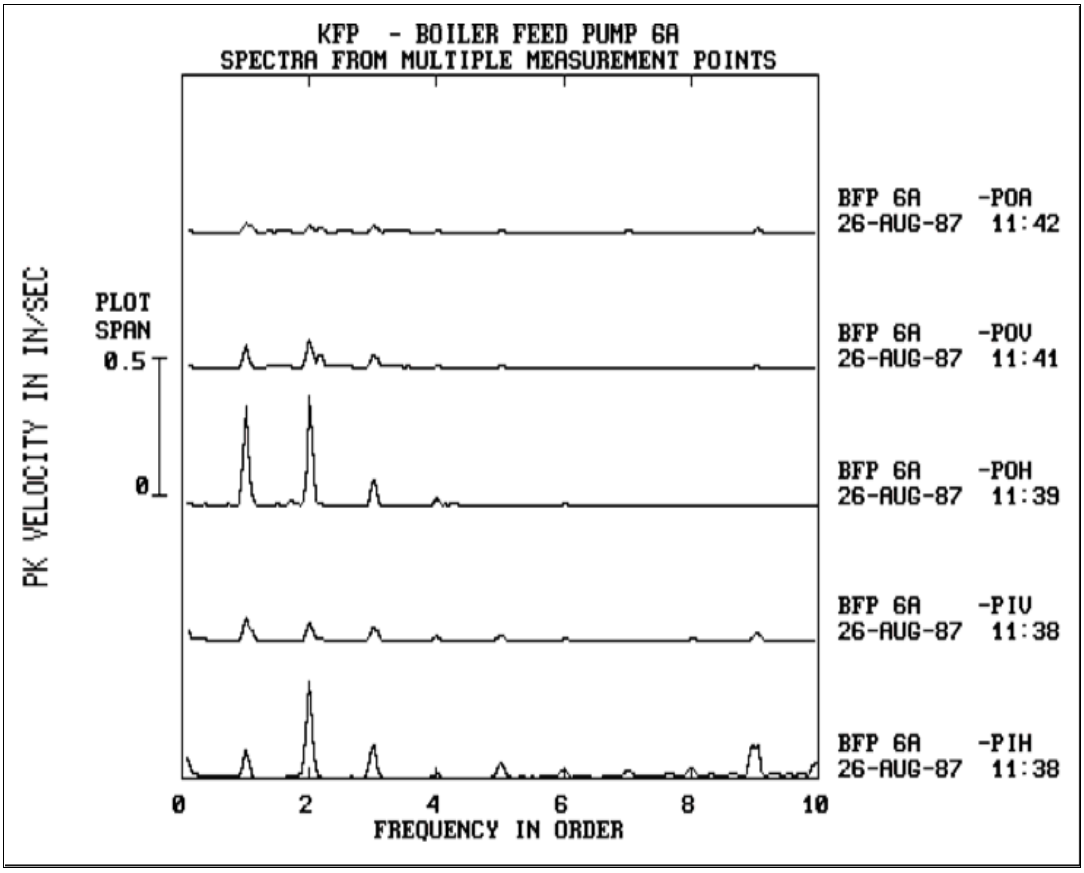


Figure 20

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

The rise of the 2xTS peak appears in this eight-month span of data taken in the pump outboard horizontal direction. The peak at 1xTS has changed very little over this time. (See Figure 21) As this bearing has worn, the internal clearance has increased, thereby allowing looseness to appear at 2xTS.

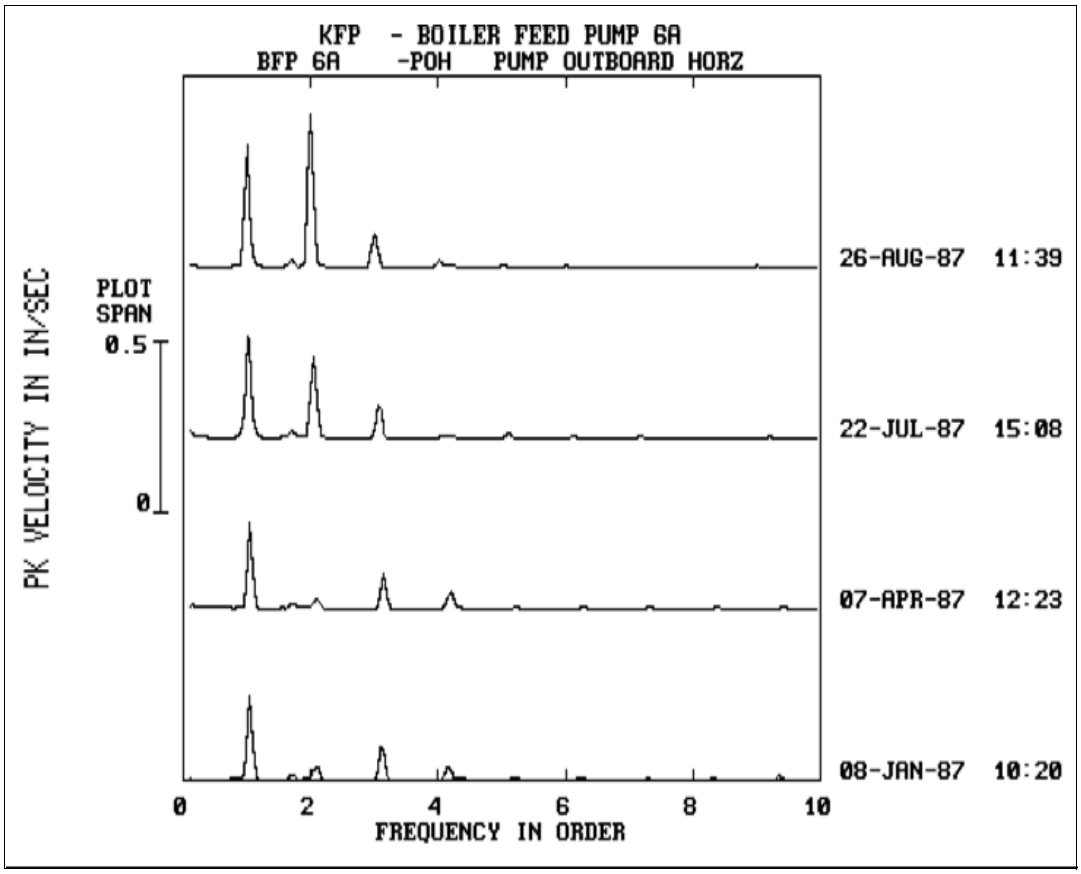


Figure 21

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Resonance

Section 10

Objectives

- Define Resonance
- Determine some causes of resonance problem
- Understand the effects of Resonance on machinery
- Establish some corrective actions

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Resonance

Every mechanical structure has at least one characteristic frequency (and sometimes more than one) called the **critical** or **resonant** frequency. When excited by an external force, the mechanical structure tends to vibrate at its resonant frequency. Less damping occurs at the resonant frequency than at other frequencies. Therefore, vibration occurring at this frequency becomes amplified. Higher levels of vibration may be observed at a machine's resonant frequency than at other frequencies. These vibration levels, however, decrease over a machine's operating lifetime.

Striking a bell or a tuning fork causes it to ring at its resonant frequency. Likewise, a machine may ring at its resonant frequency when a force such as misalignment, unbalance, bearing defects, etc. approximate that frequency. Therefore, an impact test to determine a machine's resonant frequency is sometimes called a ring test.

Stiffness, mass, and damping combine to determine a machine's resonant frequency. Changing any one of these three factors modifies the machine's resonant frequency. In turn, this alteration may help solve a resonance problem on the machine.

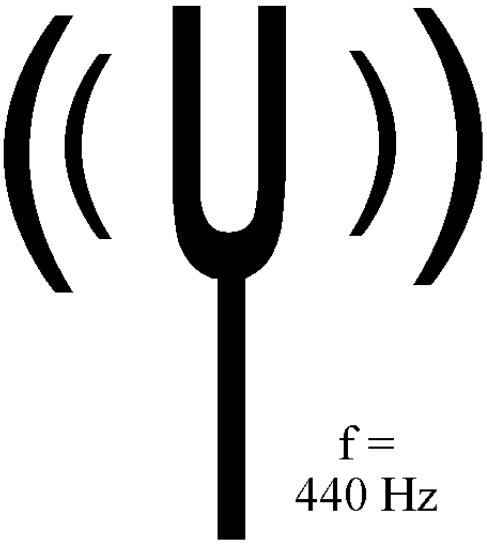


Figure 1

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Bells, tuning forks, and the strings of stringed instruments are designed to ring or resonate at particular frequencies. For example, the tuning fork in Figure 1 rings at a frequency of 440 Hz. When monitored on an oscilloscope, this frequency appears as a pure sine wave (Figure 2). In the frequency domain, the spectrum associated with this signal has only one component (Figure 3). The signal, however, decays completely (Figure 4) unless the bell is struck again. Monitor mode would show the signal decaying as the energy subsided (Figure 5.)

Figure 2

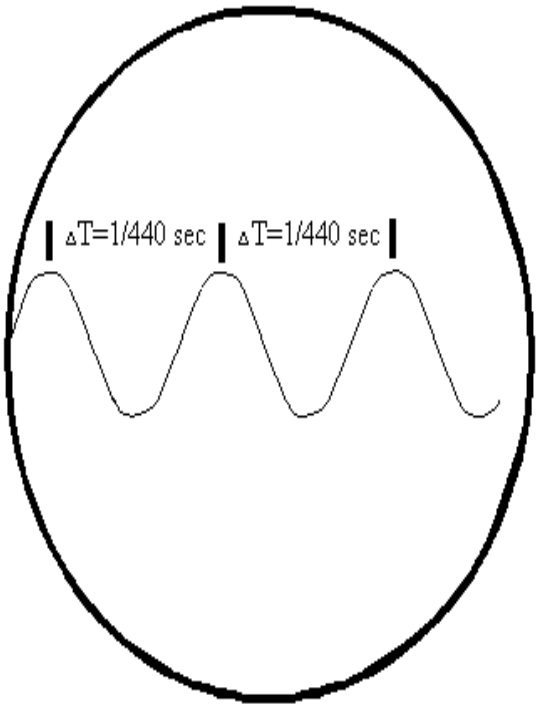
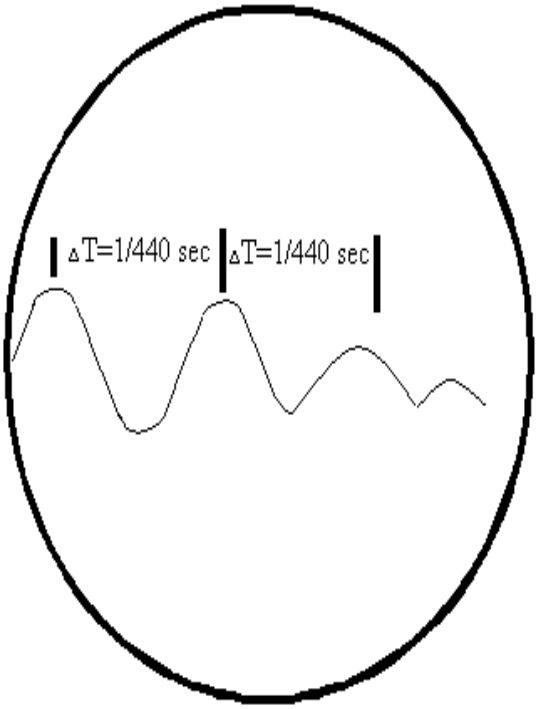


Figure 5



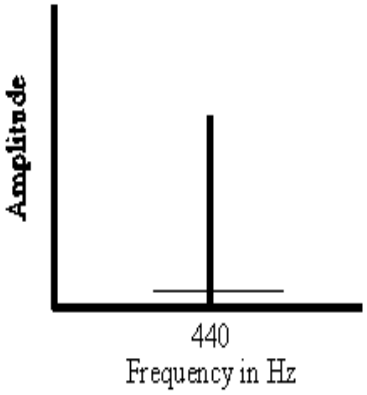


Figure 3

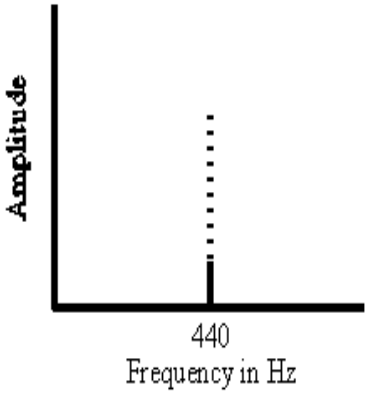


Figure 4

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Shafts also have resonant frequencies. Unlike musical instruments, they were not designed with the goal of a single resonant frequency. Most shafts, therefore, have several resonant frequencies.

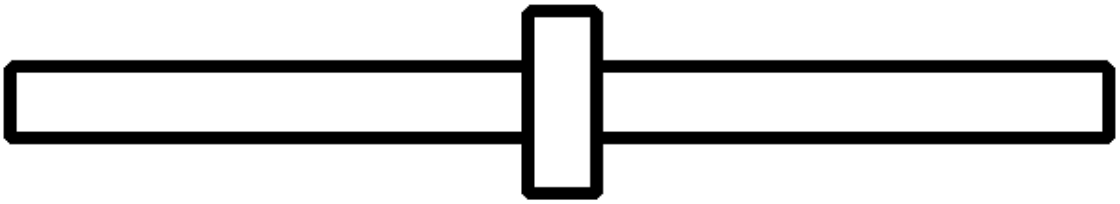


Figure 6

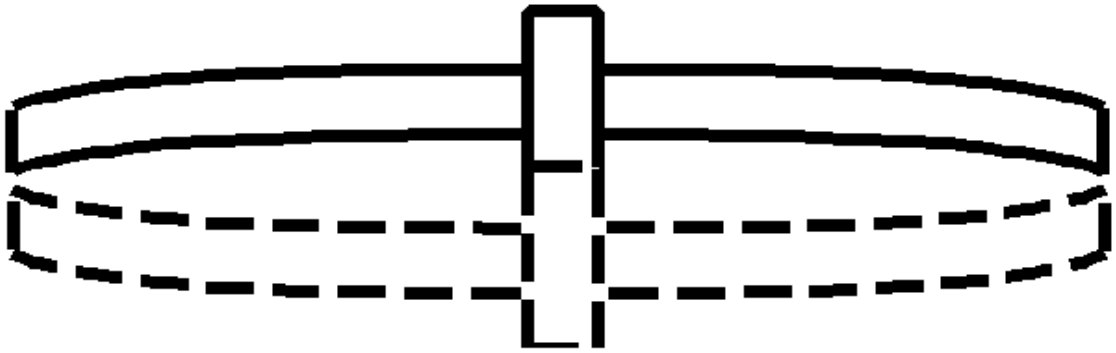


Figure 7

A shaft with several resonant frequencies should rarely be of concern on a machine that operates at only one speed. Make certain that the shaft's operational frequency is not within 20 percent of a resonant frequency to avoid exciting that frequency.

Other sources of vibration, can also excite other resonant frequencies. For example, on a given machine, the second order of vibration on a misaligned shaft may excite a shaft resonance.

When the shaft critical Frequency is excited, the shaft actually bends. The critical frequencies are the rotational frequencies of the shaft that excite system resonances(s).

 Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

.

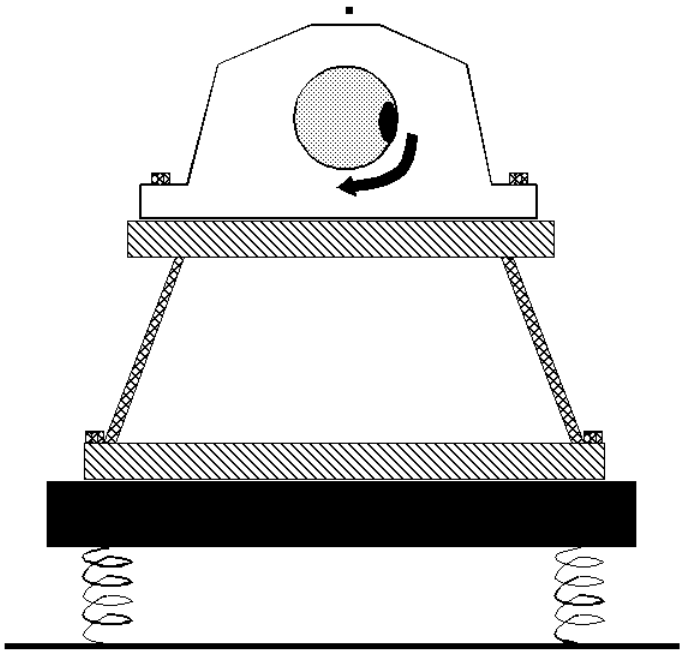


Figure 8

Many times the shaft rotation excites another part of the machine. When the heavy spot (see Figure 8) rotates to the bottom of the shaft, it tends to compress the entire unit. The stiffness of the system causes the entire unit to recoil, like a spring, when the heavy spot begins to rotate back up to the top of the shaft.

The rotation of the shaft may excite the resonant frequency of the pedestal or some other nonrotating part of the mechanical system. The shaft rotation must occur at a frequency that feeds the vibration of the nonrotating part.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

As a machine starts up, its frequency of rotation increases up to its operating speed. If the machine normally operates above resonance, the amplitude increases as the frequency increases until it reaches resonance. The amplitude then decreases after the rotational frequency passes resonance until it reaches a constant value - free space amplitude. A typical plot of amplitude vs. frequency appears in Figure 9.

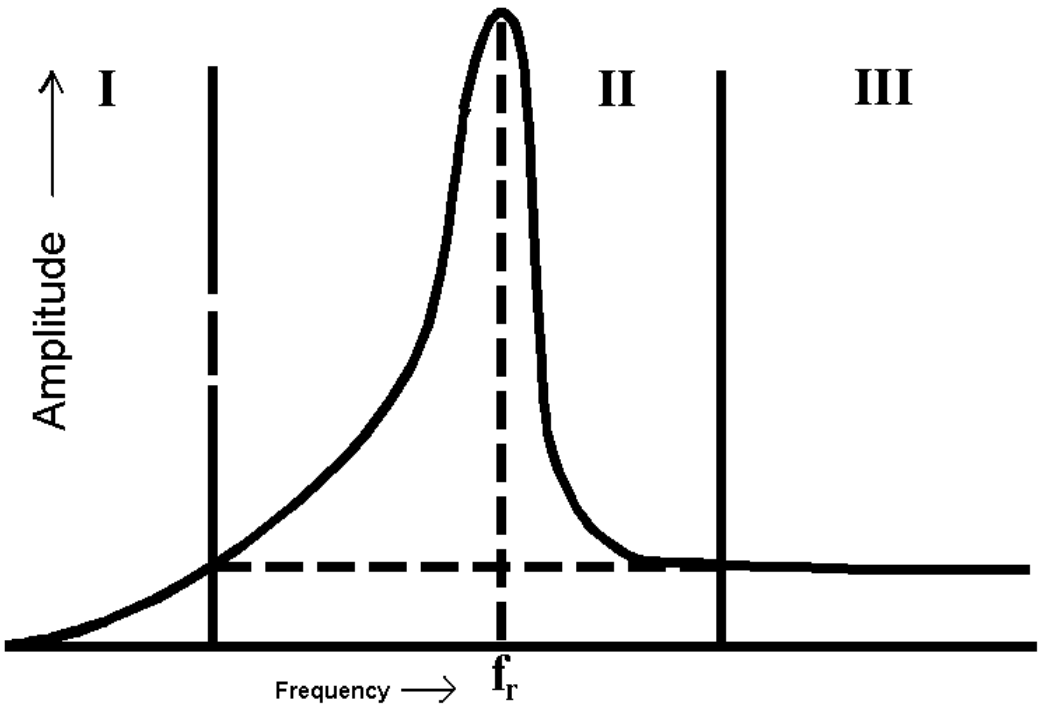


Figure 9

I Below resonance, the shaft's high vibration spot follows the heavy spot very closely. Shaft vibration is at 12 o'clock; heavy spot is also at 12 o'clock.

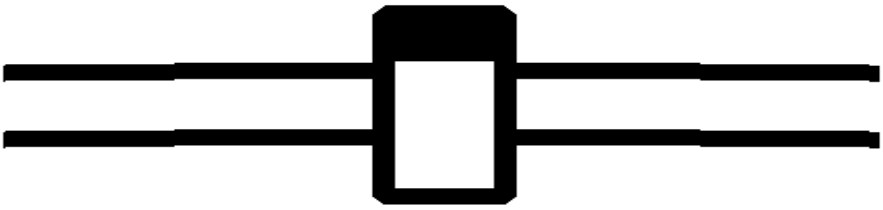


Figure 10

 Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

II Operating at the resonant frequency, the shaft vibration follows the heavy spot by 90°. Shaft vibration is at 12 o'clock; heavy spot is at 3 o'clock (if shaft vibration is clockwise). Therefore, the heavy spot has no tendency to move the shaft vertically at the instant shown. When the heavy spot rotates to 6 o'clock, the shaft vibration is at 3 o'clock, so the shaft has no tendency to resist the heavy spot. (See Figure 11) Thus, the heavy spot feeds the vibration.

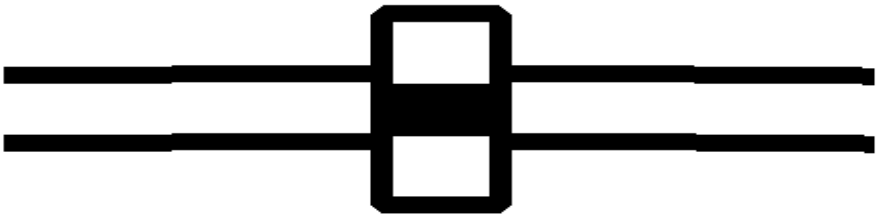


Figure 11

III Above resonance, the high vibration is on the opposite side of the shaft from the heavy spot. The shaft vibration stabilizes at a constant level determined by the unbalance in the shaft. (See Figure 12)

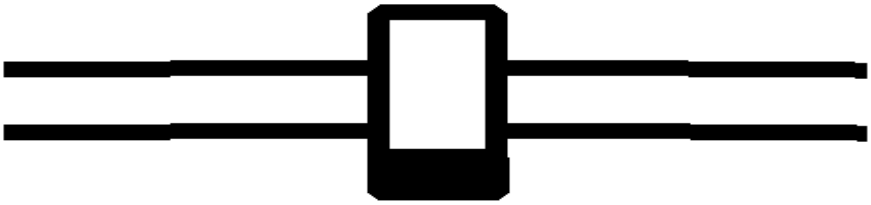


Figure 12

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Resonance

- Resonance has become more of a problem in industry in recent years than it was in the past because:

  > Equipment is now being built that runs closer to Resonance

  > Equipment is being built lighter/cheaper

  > Machines are being run at higher speeds without considering the natural resonant frequency of the equipment.

- To avoid exciting resonant frequencies, keep away from that frequency by at least 10 to 20 percent.

- Variable speed machines are more likely to be a candidate to develop a problem with resonance. Running at different speeds will increase the possibility that one or more of the run speeds will match a resonant frequency and lead to the premature failure of that equipment.

- Turning speed is not the only frequency that can cause a resonance problem.

  Harmonics of turning speed and system component frequencies may match-up with the natural frequency of that machine and lead to resonance problems/failures.

- When the system is below resonance the relationship in degrees between the heavy spot and the vibratory spot is usually closely related, about the same angle.

- When the system is running at resonance the relationship in degrees between the heavy spot and the vibratory spot is that the vibratory spot will follow the heavy spot by 90 degrees

- When the system is running above resonance the relationship is 180 degree phase shift + or **-** 30 degrees

- Three ways to change the resonant frequency:

  Add Mass - lowers Resonant frequency

  Add Stiffness - raises Resonant frequency

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Damping - Dampens vibration to keep it from being a destructive force, absorbs the energy

- The averaging mode used to perform a machine ***Running Bump Test***:

  Negative Linear Averaging -The ***Rule of Thumb*** ratio applied to draw attention to the possibility of having a Resonance problem when comparing radial readings:

  >Ratio of 3 to 1 or more (Horizontal to Vertical), suspect Resonance

- Impact Test (Bump Test) is performed to determine the system resonance on a static machine

- During a coast down, monitor peak and phase to help verify a resonant frequency.

- View the waveform in G's when performing a bump test.

  >Acceleration shows the Impacting better

- Resonance may appear anywhere in the spectrum. It can appear as sub-synchronous, synchronous, or non-synchronous energy.

  >Resonance, unlike most other forcing functions or faults is not frequency bound. A resonant frequency may appear at any frequency in the spectrum.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Case History #1 - Reactor Fan #7

Vibration was very directional on this machine. Notice the difference when comparing the Horizontal to Vertical amplitudes. This is one characteristic of Resonance. A multiple point plot of that data is displayed in Figure 13.

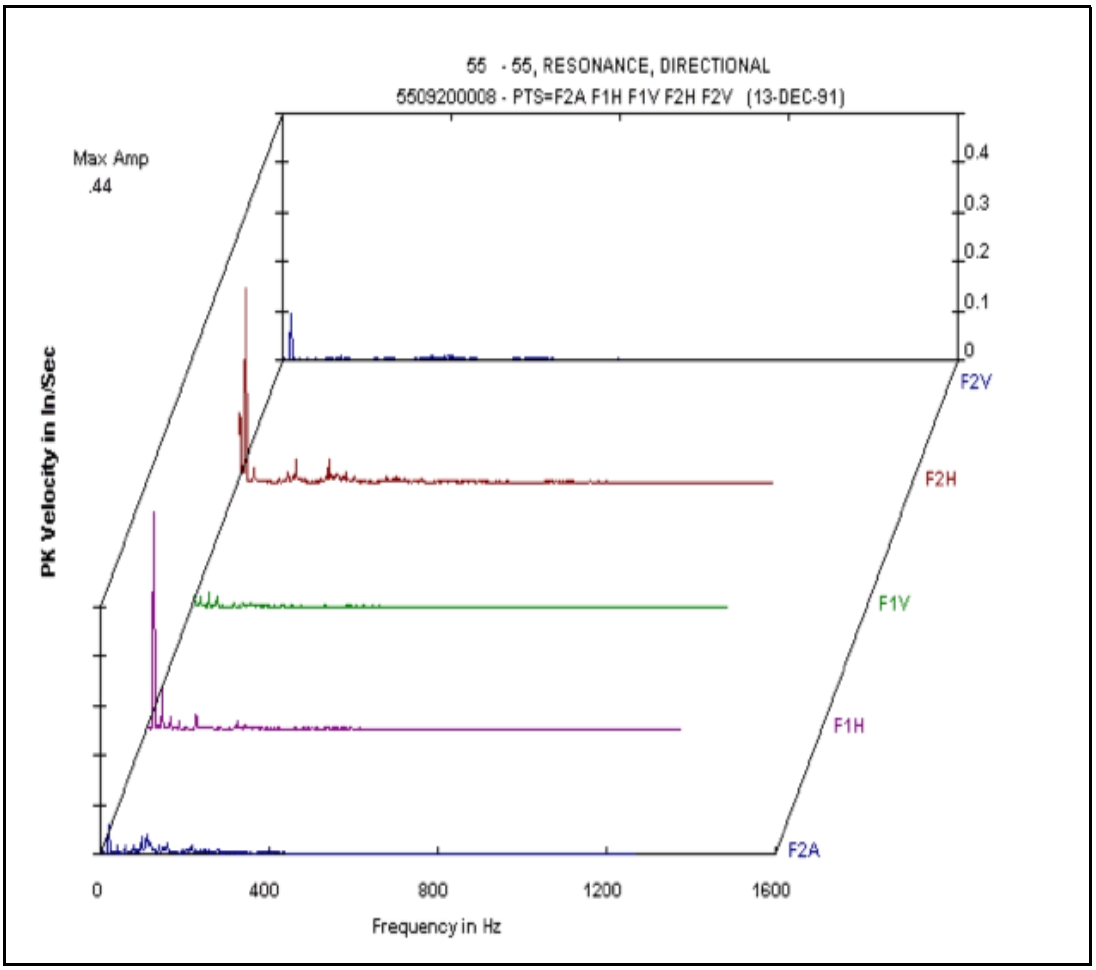


Figure 13

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Figure 14 is a single full scale plot of the fan inboard horizontal measurement point. The overall level of vibration is 0.2347 in/sec.

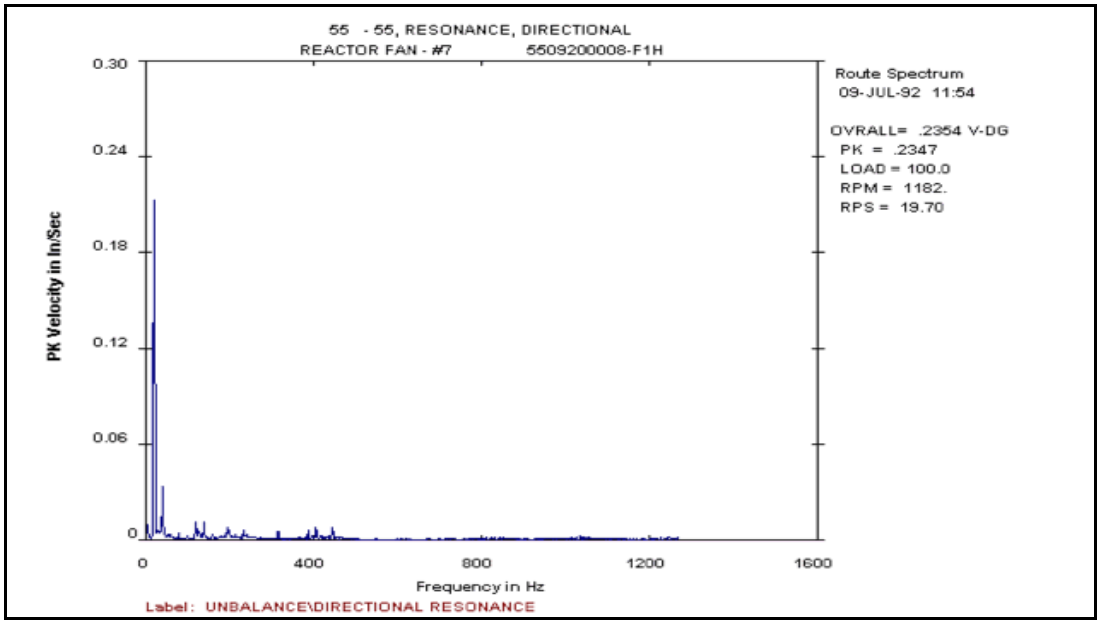


Figure 14

Examine the fan inboard vertical point in Figure 15. The overall vibration is considerably less at 0.04449 in/sec. This is more than a 5 to 1 ratio. There is possible resonance when there is a 3 to 1 ratio or more in amplitudes when comparing horizontal to vertical directions.

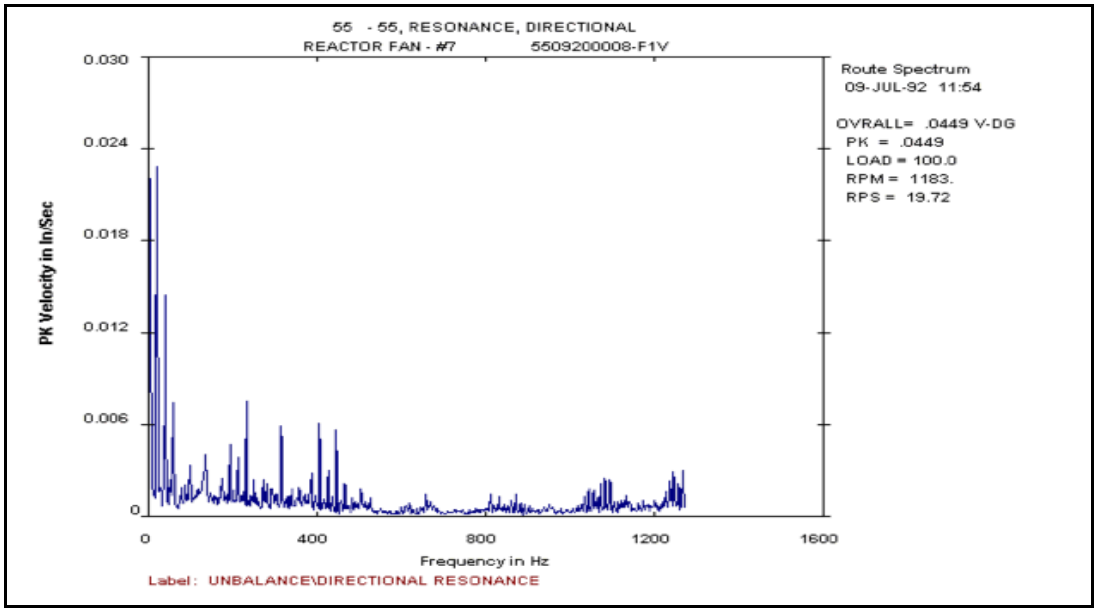


Figure 15

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

There is even a greater difference on the outboard end of the machine. Figure 16 is the fan outboard horizontal measurement point. Consider the amplitude on this point, 0.54 in/sec.

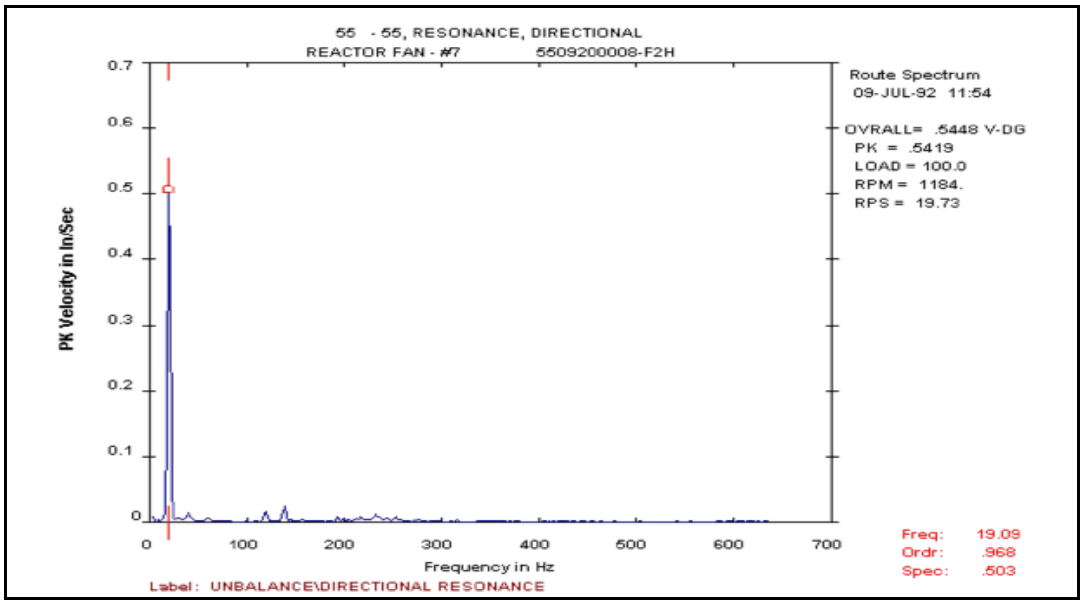


Figure 16

Compare the fan outboard horizontal (Figure 17) with the fan outboard vertical point. (Figure 18). The overall amplitude is 0.5419 in/sec. on the Horizontal direction point.

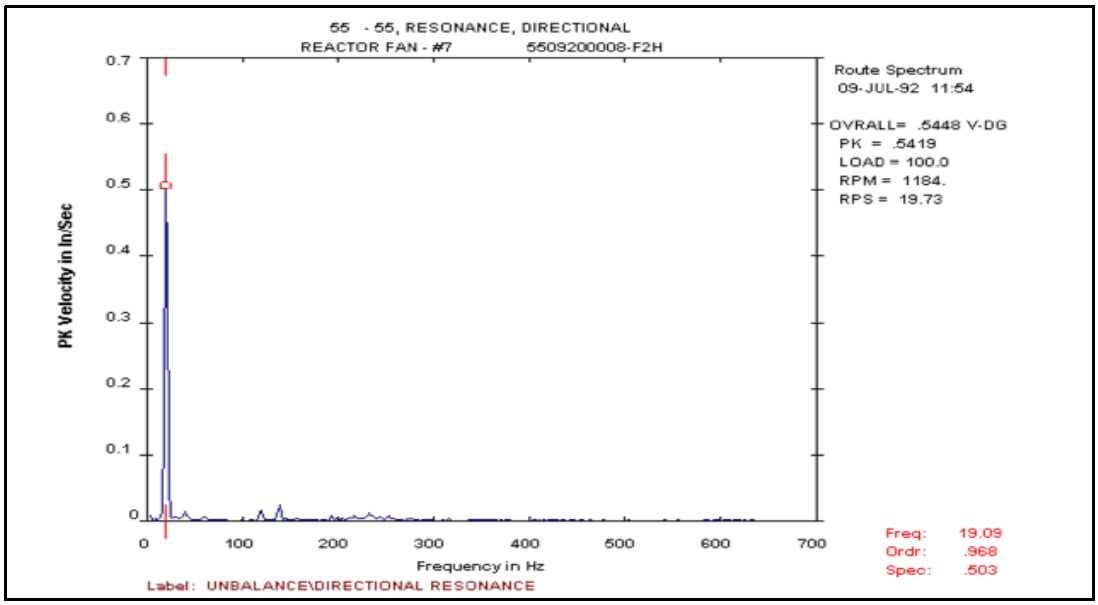


Figure 17

 Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Consider the overall amplitude of the F2V measurement point.

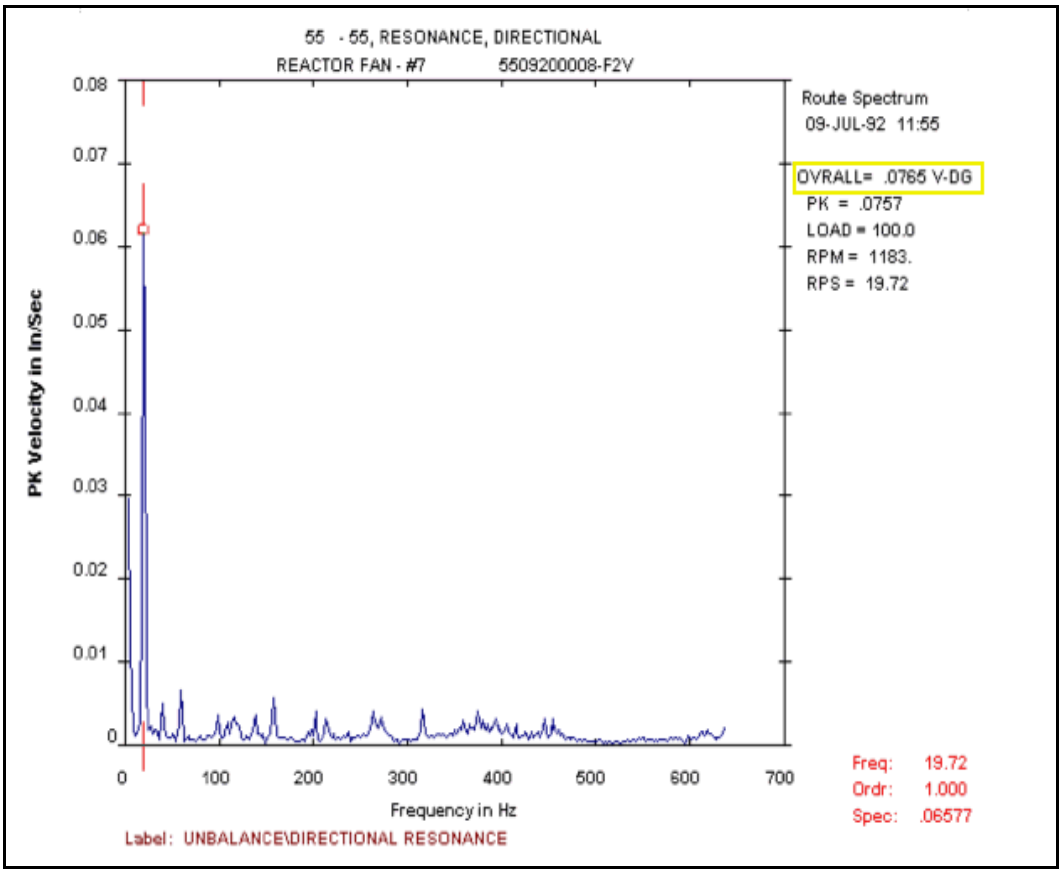


Figure 18

- The ratio of the **overall** energy is greater than 7:1.
- The ratio of the amplitude of the respective turning speed peaks is greater than 7.6:1
- Resonance is strongly suspected.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Case History #2 - DAF Pressure Pump

In this case a pump resonance was driving the vibration amplitude at almost 3x run speed to a very high level. Figure 19 is the multiple point plot for this machine. Notice the amplitudes in the horizontal, vertical, and axial directions.

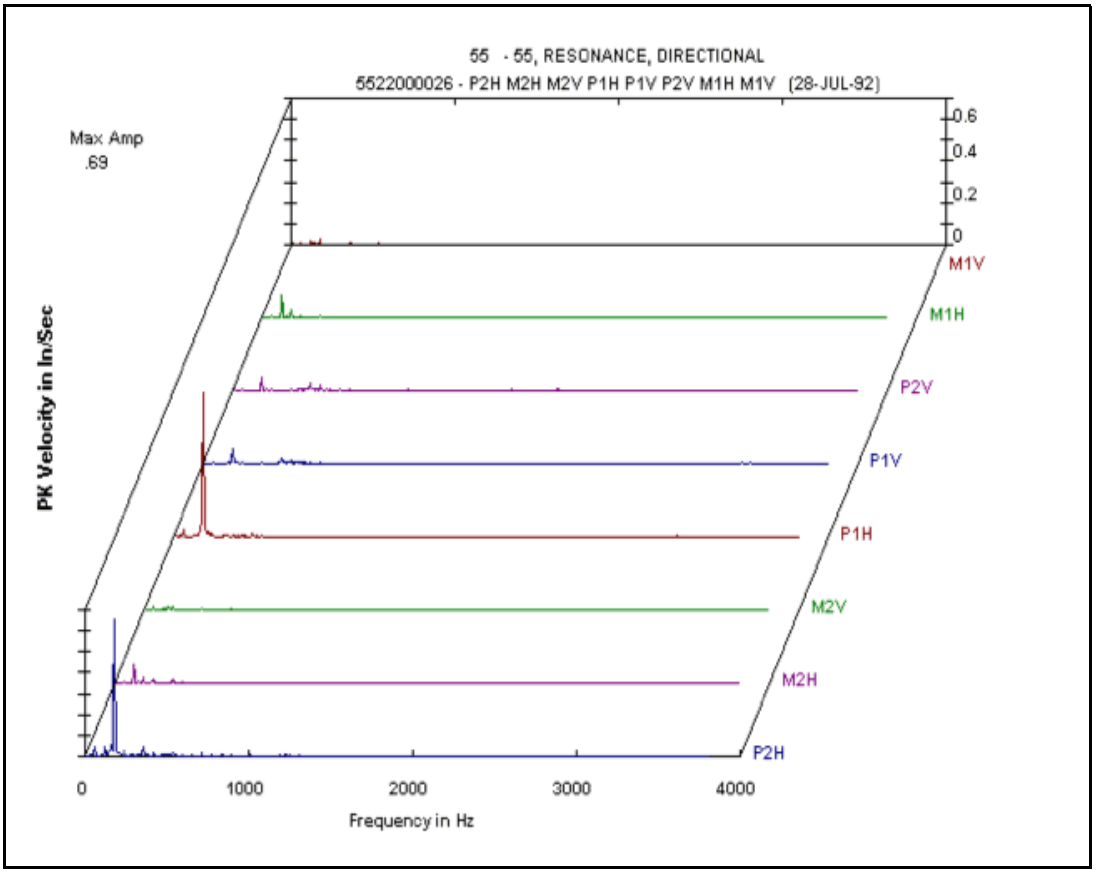


Figure 19

The amplitudes are highest on the pump points and in the Horizontal direction.

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

A full scale plot of the PIH point is shown in Figure 20. Examine the amplitude in the horizontal direction at about 3x turning speed (0.727 in/sec.)

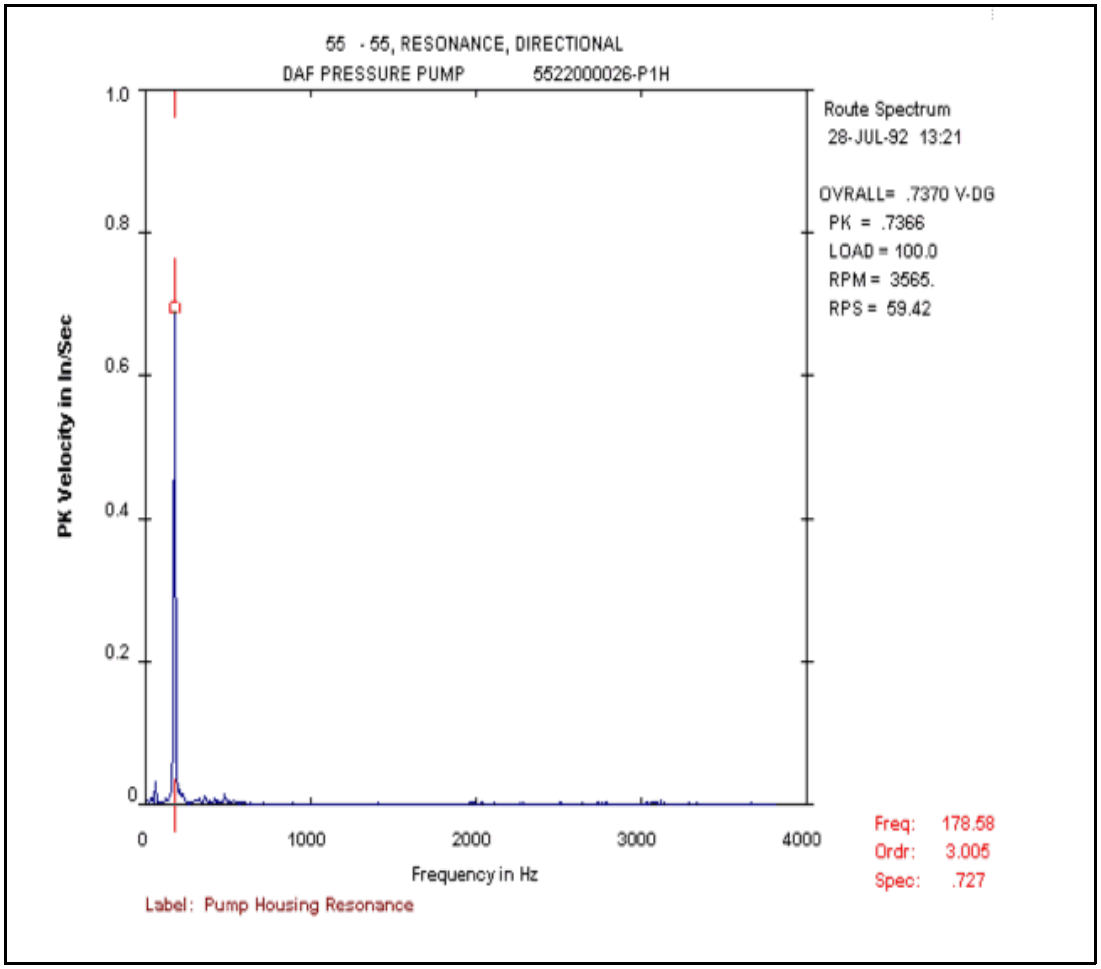


Figure 20

Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06

Compare the amplitude of the horizontal direction (Figure 20) with the amplitude of the vertical direction. (Figure 21) At 0.0944 in/sec., this is a 8 to 1 ratio. Suspect resonance.

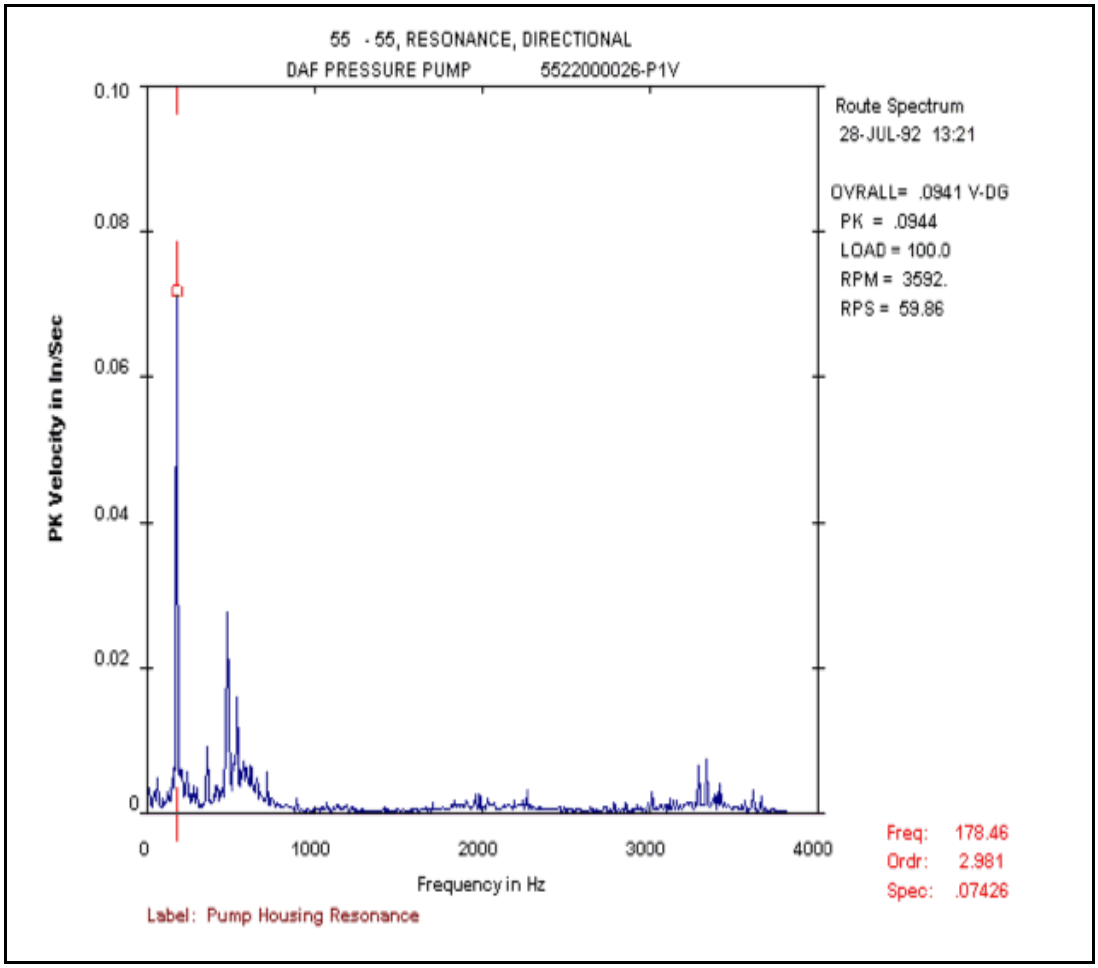


Figure 21

 Copyright 2006, Emerson Process Managment All rights reserved. Rev 04/06